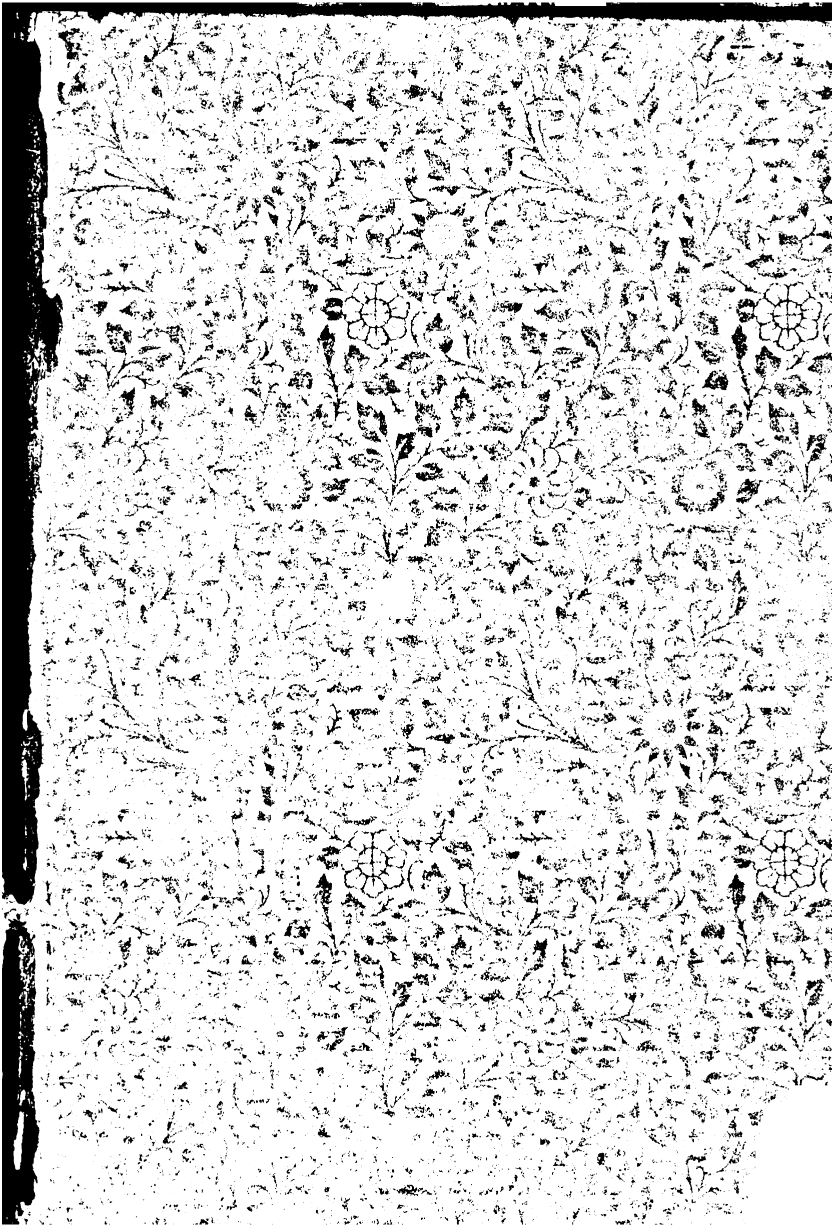

A

POCKET MANUAL

FOR

ENGINEERS.

EDITED BY

illsmith

JOHN W. HILL,

MECHANICAL ENGINEER,

Member American Society of Civil Engineers,

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PREFACE.

The following pages contain a limited number of facts, data and formulæ, together with several practical illustrations in Steam and Mechanical Engineering, which, it is hoped, will be of value to the profession and others.

Before parting with the work, the author desires to thank Mr. Harris for his liberality in permitting the extent and cost of the Manual to far exceed the original estimate, Messrs. Robert Clarke & Co. for their excellent composition and press work, and his office assistants, Messrs. Sims and Anderson, for their careful proof reading of the pages.

J. W. H.

Cincinnati, May 1, 1883.

CONTENTS.

MENSURATION.

CIRCLE.

Diam. $\times$ 3.1416 = *circumference.*
Diam.^{2} $\times$ .7854 = *area.*
Circum. $\times$ .31831 = *diameter.*

SPHERE.

Diam. $\times$ circumference = *convex surface.*
Diam^3 $\times$ .5236 = *solid contents.*
Desired convex surface of a sphere 2″ diam.

$$2 \times 6.2832 = 12.5664 \textit{ sq. ins.}$$

Desired solid contents of same sphere.

$$2^3 \times .5236 = 4.1888 \textit{ cu. ins.}$$

SPHERICAL SEGMENT.

To Find Solid Contents:

Let R = radius of base or plane surface parallel to axis, and h = height of segment or perpendicular distance from plane surface to apex of segment, then

$$\overline{3\,R^2 + h^2} \times h \times .5236 = \textit{solid contents.}$$

Desired solid contents of a spherical segment having a diam. of base 8″ = 2 R; and a height 2″ = h

$$\overline{3 \times 4^2 + 2^2} \times 2 \times .5236 = 54.45 \textit{ cu. ins.}$$

To Compute the Convex Surface of a Spherical Segment:

Let c = circum. of whole sphere, then

$$c \times h = \textit{convex surface.}$$

Desired convex surface of spherical segment: when the height h = 2″ and circumference c = 37.7″.

$$37.7 \times 2 = 75.4 \textit{ sq. ins.}$$

SPHERICAL ZONE.

To Compute Convex Surface:

$$c \times h = \textit{convex surface.}$$

Desired convex surface of zone: where the height h = 4″ and diam. = 12″.

$$12 \times 3.1416 \times 4 = 150.79 \textit{ sq. ins.}$$

To Compute Solid Contents:

Let R = radius of one plane surface: r = radius of opposite plane surface, and h = height perpendicular to plane surfaces. Then

$$\overline{R^2 + r^2 + .33\,h^2} \times h \times 1.5708 = \textit{solid contents.}$$

Desired volume of spherical zone: where the diam. of one plane surface is 8″ and diam. of opposite plane surface 6″; height 4″.

$$4^2 + 3^2 + (.33 \times 4^2) \times 4 \times 1.5708 = 190.255 \textit{ cu. ins.}$$

CONE.

To Compute Convex Surface:

Let c = circum. of base and h = *slant height* or side of cone, then

$$\frac{c \times h}{2} = \textit{convex surface.}$$

Desired convex surface of cone having a diameter of base 4″ and slant height 6″.

$$\frac{12.5664 \times 6}{2} = 37.7 \textit{ sq. ins.}$$

To Compute Solid Contents:

Let A = area of base: and h' = *perpendicular height*, then

$$\frac{A \times h'}{3} = \textit{solid contents.}$$

Desired volume of above cone:

$$\frac{(4^2 \times .7854) \times 5.6569}{3} = 23.6956 \textit{ cu. ins.}$$

NOTE.—The ratio of the solid contents of a pyramid or cone to a prism or cylinder having same area of base and perpendicular height, is as 1 : 3, and the ratio of the solid contents of a cone to a hemisphere having same area of base and perpendicular height, is as 1 : 2.

ELLIPSE.

To Compute the Area:

Let D = long diameter, and d, short diam., then

$$D \times d \times .7854 = \textit{area.}$$

Desired area of ellipse having a long diameter of 12″ and short diam. of 5″.

$$12 \times 5 \times .7854 = 47.124 \textit{ sq. ins.}$$

To Compute the Circumference or Perimeter:

The following formula is proposed by Mr. John C. Trautwine as being approximately correct to .001 of perimeter.

Let D = long diameter; d = short diameter; and a = constant as per table, then

$$3.1416\sqrt{\left(\frac{D^2 + d^2}{2}\right) - \frac{(D-d)^2}{a^*}} = \textit{circumference.}$$

The value of "a" depends upon the ratio of D to d. The values are given by Mr Trautwine as per table.

Ratio..	5	6	7	8	9	10	12	14	16	18	20
Constant (a)	8.8	9	9.2	9.3	9.35	9.4	9.5	9.6	9.68	9.75	9.8

* For ratio of less than 5 use 8.8.

SECTOR OF A CIRCLE.

To Compute the Area:

Let K = degrees of arc comprised in the sector, and A = area of whole circle, of which the sector is a part; then

$$\frac{K \times A}{360} = \textit{area of sector.}$$

Desired area of sector: where $K = 60$ degrees and area of whole circle 201.06 square inches.

$$\frac{60 \times 201.06}{360} = 33.51 \textit{ sq. ins.}$$

Or let b = length of arc, and r = radius; then

$$\frac{b \times r}{2} = \textit{area.}$$

Desired area of sector of circle: having a length of arc 8.3776″ and radius 8″.

$$\frac{8.3776 \times 8}{2} = 33.5104 \textit{ sq. ins.}$$

SEGMENT OF A CIRCLE.

To Compute Area:

From the area of the sector subtract the area of triangle formed by the chord of the segment, and the radii of the arc.

Let R = radius of arc: c = chord of segment; and h = versed sine: or height of segment; then

$$\frac{(R - h) \times c}{2} = \textit{area of triangle.}$$

Desired area of segment: area of sector = 33.5104 *sq. ins.*; $R = 8''$; $c = 8''$; and $h = 1.0718''$; then

$$33.5104 - \frac{(8 - 1.0718) \times 8}{2} = 5.7976 \textit{ sq. ins.}$$

PRISMOID.

A prismoid is a solid bounded by six plane surfaces, two of which are parallel. A frustum of a quadrangular pyramid is a prismoid.*

To Compute Solid Contents:

Let A = area of one parallel surface: A' = area of opposite parallel surface: a = area of surface at mid-depth parallel to A and A'; and h = depth or perpendicular distance from A to A'; then

$$\frac{(A + A' + 4\,a) \times h}{6} = \textit{solid contents.}$$

Desired the capacity of a reservoir of rectangular plan, the upper surface of which measures 115.04′ × 179.62′ = 20663.48′; the lower surface measures 112.11′ × 176.87′ = 19828.89′; the surface at mid-depth 113.575′ × 178.245′ = 20244.176′; and depth 7.0833′; then

$$\frac{\overline{20663.48 + 19828.89 + (4 \times 20244.176)} \times 7.0833}{6} = 143{,}400.315 \textit{ cu. ft.}$$

* This formula will apply to prisms, pyramids, cones, wedges, and to all solids having two parallel surfaces, and united by plane or curved surfaces upon which a straight line may be drawn from one parallel surface to the other, and which shall everywhere coincide with the surface upon which it is drawn.

CIRCUMFERENCES AND AREAS OF CIRCLES.

Diam. Inches.	Circum Inches.	Area Sq In.	Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area Sq. In.
1-64	.049087	.00019	¼	7.06858	3.9761	9-16	17.4751	24.301
1-32	.098175	.00077	5-16	7.26493	4.2000	⅝	17.6715	24.850
3-64	.147262	.00173	⅜	7.46128	4.4301	11-16	17.8678	25.406
1-16	.196350	.00307	7-16	7.65763	4.6664	¾	18.0642	25.967
3-32	.294524	.00690	½	7.85398	4.9087	13-16	18.2605	26.535
⅛	.392699	.01227	9-16	8.05033	5.1572	⅞	18.4569	27.109
5-32	.490874	.01917	⅝	8.24668	5.4119	15-16	18.6532	27.688
3-16	.589049	.02761	11-16	8.44303	5.6727	6.	18.8496	28.274
7-32	.687223	.03758	¾	8.63938	5.9396	⅛	19.2423	29.465
¼	.785398	.04909	13-16	8.83573	6.2126	¼	19.6350	30.680
9-32	.883573	.06213	⅞	9.03208	6.4918	⅜	20.0277	31.919
5-16	.981748	.07670	15-16	9.22843	6.7771	½	20.4204	33.183
11-32	1.07992	.09281	3.	9.42478	7.0686	⅝	20.8131	34.472
⅜	1.17810	.11045	1-16	9.62113	7.3662	¾	21.2058	35.785
13-32	1.27627	.12962	⅛	9.81748	7.6699	⅞	21.5984	37.122
7-16	1.37445	.15033	3-16	10.0138	7.9798	7.	21.9911	38.485
15-32	1.47262	.17257	¼	10.2102	8.2958	⅛	22.3838	39.871
½	1.57080	.19635	5-16	10.4065	8.6179	¼	22.7765	41.282
17-32	1.66897	.22166	⅜	10.6029	8.9462	⅜	23.1692	42.718
9-16	1.76715	.24850	7-16	10.7992	9.2806	½	23.5619	44.179
19-32	1.86532	.27688	½	10.9956	9.6211	⅝	23.9546	45.664
⅝	1.96350	.30680	9-16	11.1919	9.9678	¾	24.3473	47.173
21-32	2.06167	.33824	⅝	11.3883	10.321	⅞	24.7400	48.707
11-16	2.15984	.37122	11-16	11.5846	10.680	8.	25.1327	50.265
23-32	2.25802	.40574	¾	11.7810	11.045	⅛	25.5254	51.849
¾	2.35619	.44179	13-16	11.9773	11.416	¼	25.9181	53.456
25-32	2.45437	.47937	⅞	12.1737	11.793	⅜	26.3108	55.088
13-16	2.55254	.51849	15-16	12.3700	12.177	½	26.7035	56.745
27-32	2.65072	.55914	4.	12.5664	12.566	⅝	27.0962	58.426
⅞	2.74889	.60132	1-16	12.7627	12.962	¾	27.4889	60.132
29-32	2.84707	.64504	⅛	12.9591	13.364	⅞	27.8816	61.862
15-16	2.94524	.69029	3-16	13.1554	13.772	9.	28.2743	63.617
31-32	3.04342	.73708	¼	13.3518	14.186	⅛	28.6670	65.397
1.	3.14159	.78540	5-16	13.5481	14.607	¼	29.0597	67.201
1-16	3.33794	.88664	⅜	13.7445	15.033	⅜	29.4524	69.029
⅛	3.53129	.99402	7-16	13.9408	15.466	½	29.8451	70.882
3-16	3.73064	1.1075	½	14.1372	15.904	⅝	30.2378	72.760
¼	3.92699	1.2272	9-16	14.3335	16.349	¾	30.6305	74.662
5-16	4.12334	1.3530	⅝	14.5299	16.800	⅞	31.0232	76.549
⅜	4.31969	1.4849	11-16	14.7262	17.257	10.	31.4159	78.540
7-16	4.51604	1.6230	¾	14.9226	17.721	¼	32.2013	82.516
½	4.71239	1.7671	13-16	15.1189	18.190	½	32.9867	86.590
9-16	4.90874	1.9175	⅞	15.3153	18.665	¾	33.7721	90.763
⅝	5.10509	2.0739	15-16	15.5116	19.147	11.	34.5575	95.033
11-16	5.30144	2.2365	5.	15.7080	19.635	¼	35.3429	99.402
¾	5.49779	2.4053	1-16	15.9043	20.129	½	36.1283	103.87
13-16	5.69414	2.5802	⅛	16.1007	20.629	¾	36.9137	108.43
⅞	5.89049	2.7612	3-16	16.2970	21.135	12.	37.6991	113.10
15-16	6.08684	2.9483	¼	16.4934	21.649	¼	38.4845	117.86
2.	6.28319	3.1416	5-16	16.6897	22.166	½	39.2699	122.72
1-16	6.47953	3.3410	⅜	16.8861	22.691	¾	40.0553	127.68
⅛	6.67588	3.5466	7-16	17.0824	23.221	13.	40.8407	132.73
3-16	6.87223	3.7583	½	17.2788	23.758	¼	41.6261	137.89

CIRCUMFERENCES AND AREAS OF CIRCLES.—Continued.

Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area Sq. In.
13. ½	42.4115	143.14	¾	84.0376	562.00	40.	125.664	1256.6
¾	43.1969	148.49	27.	84.8230	572.56	¼	126.449	1272.4
14.	43.9823	153.94	¼	85.6084	583.21	½	127.235	1288.2
¼	44.7677	159.48	½	86.3938	593.96	¾	128.020	1304.2
½	45.5531	165.13	¾	87.1792	604.81	41.	128.805	1320.3
¾	46.3385	170.87	28.	87.9646	615.75	¼	129.591	1336.4
15.	47.1239	176.71	¼	88.7500	626.80	½	130.376	1352.7
¼	47.9093	182.65	½	89.5354	637.94	¾	131.161	1369.0
½	48.6947	188.69	¾	90.3208	649.18	42.	131.947	1385.4
¾	49.4801	194.83	29.	91.1062	660.52	¼	132.732	1402.0
16.	50.2655	201.06	¼	91.8916	671.96	½	133.518	1418.6
¼	51.0509	207.39	½	92.6770	683.49	¾	134.303	1435.4
½	51.8363	213.82	¾	93.4624	695.13	43.	135.088	1452.2
¾	52.6217	220.35	30.	94.2405	706.86	¼	135.874	1469.1
17.	53.4071	226.98	¼	95.0332	718.69	½	136.659	1486.2
¼	54.1925	233.71	½	95.8186	730.62	¾	137.445	1503.3
½	54.9779	240.53	¾	96.6040	742.64	44.	138.230	1520.5
¾	55.7633	247.45	31.	97.3894	754.77	¼	139.015	1537.9
18.	56.5487	254.47	¼	98.1748	766.99	½	139.801	1555.3
¼	57.3341	261.59	½	98.9602	779.31	¾	140.586	1572.8
½	58.1195	268.80	¾	99.7456	791.73	45.	141.372	1590.4
¾	58.9049	276.12	32.	100.531	804.25	¼	142.157	1608.2
19.	59.6903	283.53	¼	101.316	816.86	½	142.942	1626.0
¼	60.4757	291.04	½	102.102	829.58	¾	143.728	1643.9
½	61.2611	298.65	¾	102.887	842.39	46.	144.514	1661.9
¾	62.0465	306.35	33.	103.673	855.30	¼	145.299	1680.0
20.	62.8319	314.16	¼	104.458	868.31	½	146.084	1698.2
¼	63.6173	322.06	½	105.243	881.41	¾	146.869	1716.5
½	64.4026	330.06	¾	106.029	894.62	47.	147.655	1734.9
¾	65.1880	338.16	34.	106.814	907.92	¼	148.440	1753.5
21.	65.9734	346.36	¼	107.600	921.32	½	149.226	1772.1
¼	66.7588	354.66	½	108.385	934.82	¾	150.011	1790.8
½	67.5442	363.05	¾	109.170	948.42	48.	150.797	1809.6
¾	68.3296	371.54	35.	109.956	962.11	¼	151.582	1828.5
22.	69.1150	380.13	¼	110.741	975.91	½	152.367	1847.5
¼	69.9004	388.82	½	111.527	989.8	¾	153.153	1866.5
½	70.6858	397.61	¾	112.312	1003.8	49.	153.938	1885.7
¾	71.4712	406.49	36.	113.097	1017.9	¼	154.723	1905.0
23.	72.2566	415.48	¼	113.883	1032.1	½	155.509	1924.4
¼	73.0420	424.56	½	114.668	1046.3	¾	156.294	1943.9
½	73.8274	433.74	¾	115.454	1060.7	50.	157.080	1963.5
¾	74.6128	443.01	37.	116.239	1075.2	¼	157.865	1983.2
24.	75.3982	452.39	¼	117.024	1089.8	½	158.650	2003.0
¼	76.1836	461.86	½	117.810	1104.5	¾	159.436	2022.8
½	76.9690	471.44	¾	118.596	1119.2	51.	160.221	2042.8
¾	77.7544	481.11	38.	119.381	1134.1	¼	161.007	2062.9
25.	78.5398	490.87	¼	120.166	1149.1	½	161.792	2083.1
¼	79.3252	500.74	½	120.952	1164.2	¾	162.578	2103.3
½	80.1106	510.71	¾	121.737	1179.3	52.	163.363	2123.7
¾	80.8960	520.77	39.	122.522	1194.6	¼	164.148	2144.2
26.	81.6814	530.93	¼	123.308	1210.0	½	164.934	2164.8
¼	82.4668	541.19	½	124.093	1225.4	¾	165.719	2185.4
½	83.2522	551.55	¾	124.879	1241.0	53.	166.504	2206.2

CIRCUMFERENCES AND AREAS OF CIRCLES.

Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area Sq. In.
53. ¼	167.290	2227.0	66. ½	208.916	3473.2	79. ¾	250.542	4995.2
½	168.075	2248.0	¾	209.701	3499.4	80.	251.327	5026.5
¾	168.861	2269.1	67.	210.487	3525.6	¼	252.113	5058.0
54.	169.646	2290.2	¼	211.272	3552.0	½	252.898	5089.6
¼	170.431	2311.5	½	212.058	3578.5	¾	253.684	5121.2
½	171.217	2332.8	¾	212.843	3605.0	81.	254.469	5153.0
¾	172.002	2354.3	68.	213.628	3631.7	¼	255.254	5184.9
55.	172.788	2375.8	¼	214.414	3658.4	½	256.040	5216.8
¼	173.573	2397.5	½	215.199	3685.3	¾	256.825	5248.9
½	174.358	2419.2	¾	215.984	3712.2	82.	257.611	5281.0
¾	175.144	2441.1	69.	216.770	3739.3	¼	258.396	5313.3
56.	175.929	2463.0	¼	217.555	3766.4	½	259.181	5345.6
¼	176.715	2485.0	½	218.341	3793.7	¾	259.967	5378.1
½	177.500	2507.2	¾	219.126	3821.0	83.	260.752	5410.6
¾	178.285	2529.4	70.	219.911	3848.5	¼	261.538	5443.3
57.	179.071	2551.8	¼	220.697	3876.0	½	262.323	5476.0
¼	179.856	2574.2	½	221.482	3903.6	¾	263.108	5508.8
½	180.642	2596.7	¾	222.268	3931.4	84.	263.894	5541.8
¾	181.427	2619.4	71.	223.053	3959.2	¼	264.679	5574.8
58.	182.212	2642.1	¼	223.838	3978.1	½	265.465	5607.9
¼	182.998	2664.9	½	224.624	4015.2	¾	266.250	5641.2
½	183.783	2687.8	¾	225.409	4043.3	85.	267.035	5674.5
¾	184.569	2710.9	72.	226.195	4071.5	¼	267.821	5707.9
59.	185.354	3734.0	¼	226.980	4099.8	½	268.606	5741.5
¼	186.139	2757.2	½	227.765	4128.2	¾	269.392	5775.1
½	186.925	2780.5	¾	228.551	4156.8	86.	270.177	5808.8
¾	187.710	2803.9	73.	229.336	4185.4	¼	270.962	5842.6
60.	188.496	2827.4	¼	230.122	4214.1	½	271.748	5876.5
¼	189.281	2851.0	½	230.907	4242.9	¾	272.533	5910.6
½	190.066	2874.8	¾	231.692	4271.8	87.	273.319	5944.7
¾	190.852	2898.6	74.	232.478	4300.8	¼	274.104	5978.9
61.	191.637	2922.5	¼	233.263	4329.9	½	274.889	6013.2
¼	192.423	2946.5	½	234.049	4359.2	¾	275.675	6047.6
½	193.208	2970.6	¾	234.834	4388.5	88.	276.460	6082.1
¾	193.993	2994.8	75.	235.619	4417.9	¼	277.246	6116.7
62.	194.779	3019.1	¼	236.405	4447.4	½	278.031	6151.4
¼	195.564	3043.5	½	237.190	4477.0	¾	278.816	6186.2
½	196.350	3068.0	¾	237.976	4506.7	89.	279.602	6221.1
¾	197.135	3092.6	76.	238.761	4536.5	¼	280.387	6256.1
63.	197.920	3117.2	¼	239.546	4566.4	½	281.173	6291.2
¼	198.706	3142.0	½	240.332	4596.3	¾	281.958	6326.4
½	199.491	3166.9	¾	241.117	4626.4	90.	282.743	6361.7
¾	200.277	3191.9	77.	241.903	4656.6	¼	283.529	6397.1
64.	201.062	3217.0	¼	242.688	4686.9	½	284.314	6432.6
¼	201.847	3242.2	½	243.473	4717.3	¾	285.100	6468.2
½	202.633	3267.5	¾	244.259	4747.8	91.	285.885	6503.9
¾	203.418	3292.8	78.	245.044	4778.4	¼	286.670	6539.7
65.	204.204	3318.3	¼	245.830	4809.0	½	287.456	6575.5
¼	204.989	3343.9	½	246.615	4839.8	¾	288.241	6611.5
½	205.774	3369.6	¾	247.400	4870.7	92.	289.027	6647.6
¾	206.560	3395.3	79.	248.186	4901.7	¼	289.812	6683.8
66.	207.345	3421.2	¼	248.971	4932.7	½	290.597	6720.1
¼	208.131	3447.2	½	249.757	4963.9	¾	291.383	6756.4

CIRCUMFERENCES AND AREAS OF CIRCLES.—Continued.

Diam. Inches.	Circum Inches.	Area Sq. In.	Diam. Inches.	Circum Inches.	Area q. In.	Diam. Inches.	Circum Inches.	Area Sq. In.
93.	292.168	6792.9	95 ½	300.022	7163.0	98.	307.876	7543.0
¼	292.954	6829.5	¾	300.807	7200.6	¼	308.661	7581.5
½	293.739	6866.1	96.	301.593	7238.2	½	309.447	7620.1
¾	294.524	6902.9	¼	302.378	7276.0	¾	310.232	7658.9
94.	295.310	6939.8	½	303.164	7313.8	99.	311.018	7697.7
¼	296.095	6976.7	¾	303.949	7351.8	¼	311.803	7736.6
½	296.881	7013.8	97.	304.734	7389.8	½	312.588	7775.6
¾	297.666	7051.0	¼	305.520	7428.0	¾	313.374	781[illegible].8
95.	298.451	7088.2	½	306.305	7466.2	100.	314.159	7854.0
¼	299.237	7125.6	¾	307.091	7504.5			

FIRST EIGHT POWERS OF FIRST TEN NUMBERS.

Powers.							
1	2	3	4	5	6	7	8
1	1	1	1	1	1	1	1
2	4	8	16	32	64	128	256
3	9	27	81	243	729	2187	6561
4	16	64	256	1024	4096	16384	65536
5	25	125	625	3125	15625	78125	390625
6	36	216	1296	7776	46656	279936	1679616
7	49	343	2401	16807	117649	823543	5764801
8	64	512	4096	32768	262144	2097152	16777216
9	81	729	6561	59049	531441	4782969	43046721
10	100	1000	10000	100000	1000000	10000000	100000000

FRACTIONS OF INCH EXPRESSED IN DECIMALS.

	Decimals.
1–64	= .015625
2–64 = 1–32	= .03125
3–64	= .046875
4–64 = 2–32 = 1–16	= .0625
6–64 = 3–32	= .09375
8–64 = 4–32 = 2–16 = 1–8	= .125
10–64 = 5–32	= .15625
12–64 = 6–32 = 3–16	= .1875
14–64 = 7–32	= .21875
16–64 = 8–32 = 4–16 = 2–8 = 1–4	= .25
18–64 = 9–32	= .28125
20–64 = 10–32 = 5–16	= .3125
22–64 = 11–32	= .34375
24–64 = 12–32 = 6–16 = 3–8	= .375
26–64 = 13–32	= .40625
28–64 = 14–32 = 7–16	= .4375
30–64 = 15–32	= .46875
32–64 = 16–32 = 8–16 = 4–8 = 2–4 = 1–2	= .5
34–64 = 17–32	= .53125
36–64 = 18–32 = 9–16	= .5625
38–64 = 19–32	= .59375
40–64 = 20–32 = 10–16 = 5–8	= .625
42–64 = 21–32	= .65625
44–64 = 22–32 = 11–16	= .6875
46–64 = 23–32	= .71875
48–64 = 24–32 = 12–16 = 6–8 = 3–4	= .75
50-64 = 25–32	= .78125
52–64 = 26–32 = 13–16	= .8125
54-64 = 27–32	= .84375
56–64 = 28–32 = 14–16 = 7–8	= 875
58–64 = 29–32	= .90625
60–64 = 30–32 = 15–16	= .9375
62–64 = 31–32	= .96875
64–64 = 32–32 = 16–16 = 8–8 = 4–4 = 2–2	= 1.00000

ENGLISH AND FRENCH MEASURES.

LINEAR MEASURE.

English.	French.	U. S. in	U. S. ft.
12 inches ....1 foot	Millimetre..	.039368	
3 feet.................. ...1 yard	Centimetre.	.393685	
5½ yds ....1 rod, perch, or pole	Decimetre ..	3 93685	
40 rods. .... .1 furlong	Metre. ..	39 .3685	
8 furlongs....................1 mile	Decametre ..	393 .685	
3 miles.....................1 league	Hectometre.	....	328 071
	Kilometre ..	....	3280 71
	Myriametre.	. . .	32807.1

SQUARE MEASURE.

English.	French.	U S. sq. in.	U. S. sq. ft.
144 sq. inches......1 sq. ft	Sq. Millimetre. ...	.001549	
9 sq. ft.. ....1 sq. yd	Sq. Centimetre....	.154988	
30¼ sq. yds......1 sq. rod	Sq. Decimetre....	15 4988	
40 sq. rods.....1 sq. rood	Sq. Metre...... ..	1549 .88	
4 roods............1 acre	Sq. Decametre...	154988.	
	Hectare..	....	107630.58
	Sq. Kilometre. ..	.. ..	10763058.

CUBIC OR SOLID MEASURE.

English.	French.—Solid and Liquid.	U. S. cub. in.	U. S. cub. ft.
	Millilitre..	.0610165	
1728 cubic in.......1 cubic foot	Centilitre..	.610165	
27 cubic feet......1 cubic yard.	Decilitre...	6 .10165	
24¾ cubic feet....1 cubic perch.	Litre. ...	61 0165	
	Decalitre..	610.165	
	Hectolitre.	....	3 53105
	Kilolitre ..	....	35 .3105
	Myriolitre.	... .	353.105

LIQUID MEASURE.

U S. Standard.	Cub. in.	British Standard.	Cub. in.
4 gills...........1 pint..	28.875	4 gills..........1 pint..	34 .6592
2 pints ....1 quart..	57 .750	2 pints ....1 quart .	69 3185
4 quarts. .....1 gallon..	231.	2 quarts.......1 pottle..	138 .637
63 gallons..1 hogshead		2 pottles.....1 gallon..	277 .274

MOMENT OF INERTIA.

The moment of inertia of a rotating body is the product of the weight, *W*, into the square of the radius of gyration, *R*, of that body.

Let W = the weight of a body,
r = the external radius,
r' = the internal radius,
I = moment of inertia.

Then for a solid sphere,

$$I = W \frac{2\,r^2}{5}$$

for a hollow sphere or spherical shell,

$$I = W \frac{2\,(r^5 - r'^5)}{5\,(r^3 - r'^3)}$$

for thin, hollow sphere,

$$I = W \frac{2\,r^2}{3}$$

for cylinder, or circular disc,

$$I = W \frac{r^2}{2}$$

for hollow cylinder, or ring,

$$I = W \frac{r^2 + r'^2}{2}$$

for thin, hollow cylinder or ring,

$$I = W \frac{r^2}{1}$$

The radius of gyration or mean radius of a rotating body, is a radius the square of which is equal to the mean of the squares of the distances of its several particles from its axis.

Using same terms as for moment of inertia, the radius of gyration, *R*, of a solid sphere is

$$R = \sqrt{\frac{2\,r^2}{5}}$$

hollow sphere whose walls are thick relative to *r*,

$$R = \sqrt{\frac{2\,(r^5 - r'^5)}{5\,(r^3 - r'^3)}}$$

hollow sphere of thin material,

$$R = \sqrt{\frac{2\,r^2}{3}}$$

cylinder, or circular disc,

$$R = \sqrt{\frac{r^2}{2}}$$

hollow cylinder of thick material, or ring,

$$R = \sqrt{\frac{r^2 + r'^2}{2}}$$

hollow cylinder of thin material, or ring,

$$R = r$$

CENTRIFUGAL FORCE.

Let R = radius of gyration of body,
W = weight, in pounds of body,
V = velocity in feet, per second, at center of gyration,
C = centrifugal force in foot pounds.

Then:

$$C = \frac{W\ V^2}{R\ 32.2}$$

In estimating the centrifugal force of a fly-wheel, the bulk of weight of which is concentrated in the rim, the centrifugal force of the rim and center or arms should be separately calculated and the two results added together for centrifugal force of the whole.

REVOLVING PENDULUM.

Many railway engineers elevate the outer rail in curves, upon the principle of the revolving pendulum: the plane of the rails being perpendicular to the axis of the pendulum (not the axis of revolution).

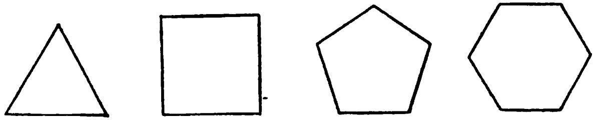
Let W = weight of railway train, or so much of train as can occupy the curve.
C = centrifugal force of train at maximum speed.
v = velocity of train in feet per second.
r = radius of curve to center of track.
h = hight above common grade, of imaginary point of suspension, of pendulum.

Then:

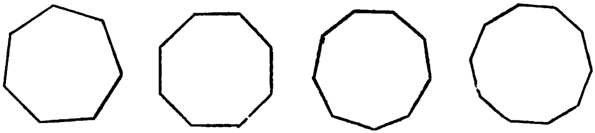
$$\frac{h}{r} = \frac{g\ r}{v^2} = \frac{W}{C} \text{ and } h = \frac{g\ r^2}{v^2}$$

Let r = sine of angle subtended by axis of revolution, and axis of pendulum, then the plane of rails should be tangent to this angle.

POLYGONS.



3. 4. 5. 6.



7. 8. 9. 10.

No. of Sides	Name of Polygon.	Areas.	Radii.	Sides.	Ang. contained between two sides.	Angle at center of circle.
3	Equilat. triangle...	.4330	.5774	1.7320	60°	120°
4	Square..	1.	.7071	1.4142	90°	90°
5	Pentagon...........	1.7205	.8507	1.1756	108°	72°
6	Hexagon..........	2.5981	1.	1.	120°	60°
7	Heptagon..........	3.6339	1.1524	.8678	128° 34.29′	51° 25 71′
8	Octagon...........	4.8284	1.3066	.7654	135°	45°
9	Nonagon...........	6.1818	1.4619	.6840	140°	40°
10	Decagon...........	7.6942	1.6180	.6180	144°	36°

Let P = number of sides or faces of polygons.
" S = side in inches of any regular polygon.
" R = radius of circumscribing circle in inches.
" R' = radius of inscribing circle in inches.
" A' = value for any given polygon, in column of areas.
" R'' = value for any given polygon, in column of radii.
" S' = value for any given polygon, in column of sides.
" A = area of polygon in sq. inches.

Then:

$$A = S^2 \times A' \text{ or } \qquad A = \frac{S \times R' \times P}{2}$$

$$R = S \times R'' \text{ and } \qquad S = R \times S'$$

SQUARE AND CUBE ROOT.

SQUARE ROOT.

RULE—Point off right to left if integer, and left to right if decimal, in orders or places of two. Ascertain highest root of first order and place to right of number as in long division. Square this root and subtract from first order. To the remainder annex the next order, double the root already obtained and place to left of this dividend; ascertain how often this divisor is contained in all but the final figure of dividend and place the quotient to right of root already obtained, and to right of the divisor. Multiply divisor by final figure in the root, and subtract as before. If the remainder after a division is negative, then take a figure for the last figure in the root one less than before.

Proceed thus until all the orders are worked.

Desired the $\sqrt{590.49}$.

```
     5,90.49(24.3
     4
     ----
 44)190
    176
    ----
483)1449
    1449
    ----
```

Desired $\sqrt{.075625}$.

```
     .07,56,25(.275*
      4
     ----
 47)356
    329
    ----
545)2725
    2725
    ----
```

* The number of decimal places in the root will always be one-half the number in the decimal the root of which is sought.

CUBE ROOT.

RULE—Point off right to left if integer, and left to right if decimal, in orders or places of three. Ascertain the highest root of the first order and place to right of number as in long division; cube the root thus found and subtract from the first order: to the remainder annex the next order, square the root already found and multiply by three for a trial divisor with two ciphers annexed. Find how often this divisor is contained in the dividend and write the result in the root.

Add together the trial divisor, three times the product of the first figure of the root by the second with one cipher annexed and the square of the second figure in the root. Multiply the sum by the last figure in the root and subtract as before.

To the remainder annex the next order, and proceed as before.

Desired the $\sqrt[3]{493039}$.

	493039(79 *cu. root.*
$7 \times 7 \times 7 =$ 343	
$7 \times 7 \times 3 =$ 14700	150039
$7 \times 9 \times 3 =$ 1890	
$9 \times 9 =$ 81	
16671	150039

Desired $\sqrt[3]{403583.419}$.

	403583.419(73.9
$7 \times 7 \times 7 =$ 343	
$7 \times 7 \times 3 =$ 14700	60583
$7 \times 3 \times 3 =$ 630	
$3 \times 3 =$ 9	
15339	46017
$73 \times 73 \times 3 =$ 1598700	14566419
$73 \times 9 \times 3 =$ 19710	
$9 \times 9 =$ 81	
1618491	14566419

Desired $\sqrt[3]{153252.632929}$.

	158252.632929(54.09*
$5 \times 5 \times 5 =$ 125	
$5 \times 5 \times 3 =$ 7500	33252
$5 \times 4 \times 3 =$ 600	
$4 \times 4 =$ 16	
8116	32464
$540 \times 540 \times 3 =$ 87480000	788632929
$540 \times 9 \times 3 =$ 145800	
$9 \times 9 =$ 81	
87625881	788632929

* When the trial divisor is greater than the dividend, write a cipher in the root, annex the next order to the dividend and proceed as before.

TABLE OF SQUARE ROOTS AND CUBE ROOTS.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
1	1.	1.	46	6.7823	3.5830	91	9.5394	4.4979
2	1.4142	1.2599	47	6.8557	3.6088	92	9.5917	4.5144
3	1.7321	1.4422	48	6.9282	3.6342	93	9.6437	4.5307
4	2.	1.5874	49	7.	3.6593	94	9.6954	4.5468
5	2.2361	1.7100	50	7.0711	3.6840	95	9.7468	4.5629
6	2.4495	1.8171	51	7.1414	3.7084	96	9.7980	4.5789
7	2.6458	1.9129	52	7.2111	3.7325	97	9.8489	4.5947
8	2.8284	2.	53	7.2801	3.7563	98	9.8995	4.6104
9	3.	2.0801	54	7.3485	3.7798	99	9.9499	4.6261
10	3.1623	2.1544	55	7.4162	3.8030	100	10.	4.6416
11	3.3166	2.2240	56	7.4833	3.8259	101	10.0499	4.6570
12	3.4641	2.2894	57	7.5498	3.8485	102	10.0995	4.6723
13	3.6056	2.3513	58	7.6158	3.8709	103	10.1489	4.6875
14	3.7417	2.4101	59	7.6811	3.8930	104	10.1980	4.7027
15	3.8730	2.4662	60	7.7460	3.9149	105	10.2470	4.7177
16	4.	2.5198	61	7.8102	3.9365	106	10.2956	4.7326
17	4.1231	2.5713	62	7.8740	3.9579	107	10.3441	4.7475
18	4.2426	2.6207	63	7.9373	3.9791	108	10.3923	4.7622
19	4.3589	2.6684	64	8.	4.	109	10.4403	4.7769
20	4.4721	2.7144	65	8.0623	4.0207	110	10.4881	4.7914
21	4.5826	2.7589	66	8.1240	4.0412	111	10.5357	4.8059
22	4.6904	2.8020	67	8.1854	4.0615	112	10.5830	4.8203
23	4.7958	2.8439	68	8.2462	4.0817	113	10.6301	4.8346
24	4.8990	2.8845	69	8.3066	4.1016	114	10.6771	4.8488
25	5.	2.9240	70	8.3666	4.1213	115	10.7238	4.8629
26	5.0990	2.9625	71	8.4261	4.1408	116	10.7703	4.8770
27	5.1962	3.	72	8.4853	4.1602	117	10.8167	4.8910
28	5.2915	3.0366	73	8.5440	4.1793	118	10.8628	4.9049
29	5.3852	3.0723	74	8.6023	4.1983	119	10.9087	4.9187
30	5.4772	3.1072	75	8.6603	4.2172	120	10.9545	4.9324
31	5.5678	3.1414	76	8.7178	4.2358	121	11.	4.9461
32	5.6569	3.1748	77	8.7750	4.2543	122	11.0454	4.9597
33	5.7446	3.2075	78	8.8318	4.2727	123	11.0905	4.9732
34	5.8310	3.2396	79	8.8882	4.2908	124	11.1355	4.9866
35	5.9161	3.2711	80	8.9443	4.3089	125	11.1803	5.
36	6.	3.3019	81	9.	4.3267	126	11.2250	5.0133
37	6.0828	3.3322	82	9.0554	4.3445	127	11.2694	5.0265
38	6.1644	3.3620	83	9.1104	4.3621	128	11.3137	5.0397
39	6.2450	3.3912	84	9.1652	4.3795	129	11.3578	5.0528
40	6.3246	3.4200	85	9.2195	4.3968	130	11.4018	5.0658
41	6.4031	3.4482	86	9.2736	4.4140	131	11.4455	5.0788
42	6.4807	3.4760	87	9.3274	4.4310	132	11.4891	5.0916
43	6.5574	3.5034	88	9.3808	4.4480	133	11.5326	5.1045
44	6.6332	3.5303	89	9.4340	4.4647	134	11.5758	5.1172
45	6.7082	3.5569	90	9.4868	4.4814	135	11.6190	5.1299

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
136	11.6619	5.1426	186	13.6382	5.7083	236	15.3623	6.1797
137	11.7047	5.1551	187	13.6748	5.7185	237	15.3948	6.1885
138	11.7473	5.1676	188	13.7113	5.7287	238	15.4272	6.1972
139	11.7898	5.1801	189	13.7477	5.7388	239	15.4596	6.2058
140	11.8322	5.1925	190	13.7840	5.7489	240	15.4919	6.2145
141	11.8743	5.2048	191	13.8203	5.7590	241	15.5242	6.2231
142	11.9164	5.2171	192	13.8564	5.7690	242	15.5563	6.2317
143	11.9583	5.2293	193	13.8924	5.7790	243	15.5885	6.2403
144	12.	5.2415	194	13.9284	5.7890	244	15.6205	6.2488
145	12.0416	5.2536	195	13.9642	5.7989	245	15.6525	6.2573
146	12.0830	5.2656	196	14.	5.8088	246	15.6844	6.2658
147	12.1244	5.2776	197	14.0357	5.8186	247	15.7162	6.2743
148	12.1655	5.2896	198	14.0712	5.8285	248	15.7480	6.2828
149	12.2066	5.3015	199	14.1067	5.8383	249	15.7797	6.2912
150	12.2474	5.3133	200	14.1421	5.8480	250	15.8114	6.2996
151	12.2882	5.3251	201	14.1774	5.8578	251	15.8430	6.3080
152	12.3288	5.3368	202	14.2127	5.8675	252	15.8745	6.3164
153	12.3693	5.3485	203	14.2478	5.8771	253	15.9060	6.3247
154	12.4097	5.3601	204	14.2829	5.8868	254	15.9374	6.3330
155	12.4499	5.3717	205	14.3178	5.8964	255	15.9687	6.3413
156	12.4900	5.3832	206	14.3527	5.9059	256	16.	6.3496
157	12.5300	5.3947	207	14.3875	5.9155	257	16.0312	6.3579
158	12.5698	5.4061	208	14.4222	5.9250	258	16.0624	6.3661
159	12.6095	5.4175	209	14.4568	5.9345	259	16.0935	6.3743
160	12.6491	5.4288	210	14.4914	5.9439	260	16.1245	6.3825
161	12.6886	5.4401	211	14.5258	5.9533	261	16.1555	6.3907
162	12.7279	5.4514	212	14.5602	5.9627	262	16.1864	6.3988
163	12.7671	5.4626	213	14.5945	5.9721	263	16.2173	6.4070
164	12.8062	5.4737	214	14.6287	5.9814	264	16.2481	6.4151
165	12.8452	5.4848	215	14.6629	5.9907	265	16.2788	6.4232
166	12.8841	5.4959	216	14.6969	6.	266	16.3095	6.4312
167	12.9228	5.5069	217	14.7309	6.0092	267	16.3401	6.4393
168	12.9615	5.5178	218	14.7648	6.0185	268	16.3707	6.4473
169	13.	5.5288	219	14.7986	6.0277	269	16.4012	6.4553
170	13.0384	5.5397	220	14.8324	6.0368	270	16.4317	6.4633
171	13.0767	5.5505	221	14.8661	6.0459	271	16.4621	6.4713
172	13.1140	5.5613	222	14.8997	6.0550	272	16.4924	6.4792
173	13.1529	5.5721	223	14.9332	6.0641	273	16.5227	6.4872
174	13.1909	5.5828	224	14.9666	6.0732	274	16.5529	6.4951
175	13.2288	5.5934	225	15.	6.0822	275	16.5831	6.5030
176	13.2665	5.6041	226	15.0333	6.0912	276	16.6132	6.5108
177	13.3041	5.6147	227	15.0665	6.1002	277	16.6433	6.5187
178	13.3417	5.6252	228	15.0997	6.1091	278	16.6733	6.5265
179	13.3791	5.6357	229	15.1327	6.1180	279	16.7033	6.5343
180	13.4164	5.6462	230	15.1658	6.1269	280	16.7332	6.5421
181	13.4536	5.6567	231	15.1987	6.1358	281	16.7631	6.5499
182	13.4907	5.6671	232	15.2315	6.1446	282	16.7929	6.5577
183	13.5277	5.6774	233	15.2643	6.1534	283	16.8226	6.5654
184	13.5647	5.6877	234	15.2971	6.1622	284	16.8523	6.5731
185	13.6015	5.6980	235	15.3297	6.1710	285	16.8819	6.5808

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
286	16.9115	6.5885	336	18.3303	6.9521	386	19.6469	7.2811
287	16.9411	6.5962	337	18.3576	6.9589	387	19.6723	7.2874
288	16.9706	6.6039	338	18.3848	6.9658	388	19.6977	7.2936
289	17.	6.6115	339	18.4120	6.9727	389	19.7231	7.2999
290	17.0294	6.6191	340	18.4391	6.9795	390	19.7484	7.3061
291	17.0587	6.6267	341	18.4662	6.9864	391	19.7737	7.3124
292	17.0880	6.6343	342	18.4932	6.9932	392	19.7990	7.3186
293	17.1172	6.6419	343	18.5203	7.	393	19.8242	7.3248
294	17.1464	6.6494	344	18.5472	7.0068	394	19.8494	7.3310
295	17.1756	6.6569	345	18.5742	7.0136	395	19.8746	7.3372
296	17.2047	6.6644	346	18.6011	7.0203	396	19.8997	7.3434
297	17.2337	6.6719	347	18.6279	7.0271	397	19.9249	7.3496
298	17.2627	6.6794	348	18.6548	7.0338	398	19.9499	7.3558
299	17.2916	6.6869	349	18.6815	7.0406	399	19.9750	7.3619
300	17.3205	6.6943	350	18.7083	7.0473	400	20.	7.3681
301	17.3494	6.7018	351	18.7350	7.0540	401	20.0250	7.3742
302	17.3781	6.7092	352	18.7617	7.0607	402	20.0499	7.3803
303	17.4069	6.7166	353	18.7883	7.0674	403	20.0749	7.3864
304	17.4356	6.7240	354	18.8149	7.0740	404	20.0998	7.3925
305	17.4642	6.7313	355	18.8414	7.0807	405	20.1246	7.3986
306	17.4929	6.7387	356	18.8680	7.0873	406	20.1494	7.4047
307	17.5214	6.7460	357	18.8944	7.0940	407	20.1742	7.4108
308	17.5499	6.7533	358	18.9209	7.1006	408	20.1990	7.4169
309	17.5784	6.7606	359	18.9473	7.1072	409	20.2237	7.4229
310	17.6068	6.7679	360	18.9737	7.1138	410	20.2485	7.4290
311	17.6352	6.7752	361	19	7.1204	411	20.2731	7.4350
312	17.6635	6.7824	362	19.0263	7.1269	412	20.2978	7.4410
313	17.6918	6.7897	363	19.0526	7.1335	413	20.3224	7.4470
314	17.7200	6.7969	364	19.0788	7.1400	414	20.3470	7.4530
315	17.7482	6.8041	365	19.1050	7.1466	415	20.3715	7.4590
316	17.7764	6.8113	366	19.1311	7.1531	416	20.3961	7.4650
317	17.8045	6.8185	367	19.1572	7.1596	417	20.4206	7.4710
318	17.8326	6.8256	368	19.1833	7.1667	418	20.4450	7.4770
319	17.8606	6.8328	369	19.2094	7.1726	419	20.4695	7.4829
320	17.8885	6.8399	370	19.2354	7.1791	420	20.4939	7.4889
321	17.9165	6.8470	371	19.2614	7.1855	421	20.5183	7.4948
322	17.9444	6.8541	372	19.2873	7.1920	422	20.5426	7.5007
323	17.9722	6.8612	373	19.3132	7.1984	423	20.5670	7.5067
324	18.	6.8683	374	19.3391	7.2048	424	20.5913	7.5126
325	18.0278	6.8753	375	19.3649	7.2112	425	20.6155	7.5185
326	18.0555	6.8824	376	19.3907	7.2177	426	20.6398	7.5244
327	18.0831	6.8894	377	19.4165	7.2240	427	20.6640	7.5302
328	18.1108	6.8964	378	19.4122	7.2304	428	20.6882	7.5361
329	18.1384	6.9034	379	19.4679	7.2368	429	20.7123	7.5420
330	18.1659	6.9104	380	19.4936	7.2432	430	20.7364	7.5478
331	18.1934	6.9174	381	19.5192	7.2495	431	20.7605	7.5537
332	18.2209	6.9244	382	19.5448	7.2558	432	20.7846	7.5595
333	18.2483	6.9313	383	19.5704	7.2622	433	20.8087	7.5654
334	18.2757	6.9382	384	19.5959	7.2685	434	20.8327	7.5712
335	18.3030	6.9451	385	19.6214	7.2748	435	20.8567	7.5770

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
436	20.8806	7.5828	486	22.0454	7.8622	536	23.1517	8.1231
437	20.9045	7.5886	487	22.0681	7.8676	537	23.1733	8.1281
438	20.9284	7.5944	488	22.0907	7.8730	538	23.1948	8.1332
439	20.9523	7.6001	489	22.1133	7.8784	539	23.2164	8.1382
440	20.9762	7.6059	490	22.1359	7.8837	540	23.2379	8.1433
441	21.	7.6117	491	22.1585	7.8891	541	23.2594	8.1483
442	21.0238	7.6174	492	22.1811	7.8944	542	23.2809	8.1533
443	21.0476	7.6232	493	22.2036	7.8998	543	23.3024	8.1583
444	21.0713	7.6289	494	22.2261	7.9051	544	23.3238	8.1633
445	21.0950	7.6346	495	22.2486	7.9105	545	23.3452	8.1683
446	21.1187	7.6403	496	22.2711	7.9158	546	23.3666	8.1733
447	21.1424	7.6460	497	22.2935	7.9211	547	23.3880	8.1783
448	21.1660	7.6517	498	22.3159	7.9264	548	23.4094	8.1833
449	21.1896	7.6574	499	22.3383	7.9317	549	23.4307	8.1882
450	21.2132	7.6631	500	22.3607	7.9370	550	23.4521	8.1932
451	21.2368	7.6688	501	22.3830	7.9423	551	23.4734	8.1982
452	21.2603	7.6744	502	22.4054	7.9476	552	23.4947	8.2031
453	21.2838	7.6801	503	22.4277	7.9528	553	23.5160	8.2081
454	21.3073	7.6857	504	22.4499	7.9581	554	23.5372	8.2130
455	21.3307	7.6914	505	22.4722	7.9634	555	23.5584	8.2180
456	21.3542	7.6970	506	22.4944	7.9686	556	23.5797	8.2229
457	21.3776	7.7026	507	22.5167	7.9739	557	23.6008	8.2278
458	21.4009	7.7082	508	22.5389	7.9791	558	23.6220	8.2327
459	21.4243	7.7138	509	22.5610	7.9843	559	23.6432	8.2377
460	21.4476	7.7194	510	22.5832	7.9896	560	23.6643	8.2426
461	21.4709	7.7250	511	22.6053	7.9948	561	23.6854	8.2475
462	21.4942	7.7306	512	22.6274	8.	562	23.7065	8.2524
463	21.5174	7.7362	513	22.6495	8.0052	563	23.7276	8.2573
464	21.5407	7.7418	514	22.6716	8.0104	564	23.7487	8.2621
465	21.5639	7.7473	515	22.6936	8.0156	565	23.7697	8.2670
466	21.5870	7.7529	516	22.7156	8.0208	566	23.7908	8.2719
467	21.6102	7.7584	517	22.7376	8.0260	567	23.8118	8.2768
468	21.6333	7.7639	518	22.7596	8.0311	568	23.8328	8.2816
469	21.6564	7.7695	519	22.7816	8.0363	569	23.8537	8.2865
470	21.6795	7.7750	520	22.8035	8.0415	570	23.8747	8.2913
471	21.7025	7.7805	521	22.8254	8.0466	571	23.8956	8.2962
472	21.7256	7.7860	522	22.8473	8.0517	572	23.9165	8.3010
473	21.7486	7.7915	523	22.8692	8.0569	573	23.9374	8.3059
474	21.7715	7.7970	524	22.8910	8.0620	574	23.9582	8.3107
475	21.7945	7.8025	525	22.9129	8.0671	575	23.9792	8.3155
476	21.8174	7.8079	526	22.9347	8.0723	576	24.	8.3203
477	21.8403	7.8134	527	22.9565	8.0774	577	24.0208	8.3251
478	21.8632	7.8188	528	22.9783	8.0825	578	24.0416	8.3300
479	21.8861	7.8243	529	23.	8.0876	579	24.0624	8.3348
480	21.9089	7.8297	530	23.0217	8.0927	580	24.0832	8.3396
481	21.9317	7.8352	531	23.0434	8.0978	581	24.1039	8.3443
482	21.9545	7.8406	532	23.0651	8.1028	582	24.1247	8.3491
483	21.9773	7.8460	533	23.0868	8.1079	583	24.1454	8.3539
484	22.	7.8514	534	23.1084	8.1130	584	24.1661	8.3587
485	22.0227	7.8568	535	23.1301	8.1180	585	24.1868	8.3634

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

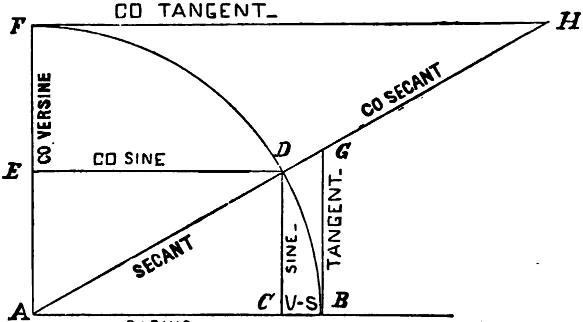
No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
586	24.2074	8.3682	636	25.2190	8.5997	686	26.1916	8.9194
587	24.2281	8.3730	637	25.2389	8.6043	687	26.2107	8.8237
588	24.2487	8.3777	638	25.2587	8.6088	688	26.2298	8.8280
589	24.2693	8.3825	639	25.2784	8.6132	689	26.2488	8.8323
590	24.2899	8.3872	640	25.2982	8.6177	690	26.2679	8.8366
591	24.3105	8.3919	641	25.3180	8.6222	691	26.2869	8.8408
592	24.3311	8.3967	642	25.3377	8.6267	692	26.3059	8.8451
593	24.3516	8.4014	643	25.3574	8.6312	693	26.3249	8.8493
594	24.3721	8.4061	644	25.3772	8.6357	694	26.3439	8.8536
595	24.3926	8.4108	645	25.3969	8.6401	695	26.3629	8.8578
596	24.4131	8.4155	646	25.4165	8.6446	696	26.3818	8.8621
597	24.4336	8.4202	647	25.4362	8.6490	697	26.4008	8.8663
598	24.4540	8.4249	648	25.4558	8.6535	698	26.4197	8.8706
599	24.4744	8.4296	649	25.4755	8.6579	699	26.4386	8.8748
600	24.4949	8.4343	650	25.4951	8.6624	700	26.4575	8.8790
601	24.5153	8.4390	651	25.5147	8.6668	701	26.4764	8.8833
602	24.5357	8.4437	652	25.5343	8.6713	702	26.4953	8.8875
603	24.5561	8.4484	653	25.5539	8.6757	703	26.5141	8.8917
604	24.5764	8.4530	654	25.5434	8.6801	704	26.5330	8.8959
605	24.5967	8.4577	655	25.5930	8.6845	705	26.5518	8.9001
606	24.6171	8.4623	656	25.6125	8.6890	706	26.5707	8.9043
607	24.6374	8.4670	657	25.6320	8.6934	707	26.5895	8.9085
608	24.6577	8.4716	658	25.6515	8.6978	708	26.6083	8.9127
609	24.6779	8.4763	659	25.6710	8.7022	709	26.6271	8.9169
610	24.6982	8.4809	660	25.6905	8.7066	710	26.6458	8.9211
611	24.7184	8.4856	661	25.7099	8.7110	711	26.6646	8.9253
612	24.7386	8.4902	662	25.7294	8.7154	712	26.6833	8.9295
613	24.7588	8.4948	663	25.7488	8.7198	713	26.7021	8.9337
614	24.7790	8.4994	664	25.7682	8.7241	714	26.7208	8.9378
615	24.7992	8.5040	665	25.7876	8.7285	715	26.7395	8.9420
616	24.8193	8.5086	666	25.8070	8.7329	716	26.7582	8.9462
617	24.8395	8.5132	667	25.8263	8.7373	717	26.7769	8.9503
618	24.8596	8.5178	668	25.8457	8.7416	718	26.7955	8.9545
619	24.8797	8.5224	669	25.8650	8.7460	719	26.8142	8.9587
620	24.8989	8.5270	670	25.8844	8.7503	720	26.8328	8.9628
621	24.9199	8.5316	671	25.9037	8.7547	721	26.8514	8.9670
622	24.9399	8.5362	672	25.9230	8.7590	722	26.8701	8.9711
623	24.9600	8.5408	673	25.9422	8.7634	723	26.8887	8.9752
624	24.9800	8.5453	674	25.9615	8.7677	724	26.9072	8.9794
625	25.	8.5499	675	25.9808	8.7721	725	26.9258	8.9835
626	25.0200	8.5544	676	26.	8.7764	726	26.9444	8.9876
627	25.0400	8.5590	677	26.0192	8.7807	727	26.9629	8.9918
628	25.0599	8.5635	678	26.0384	8.7850	728	26.9815	8.9559
629	25.0799	8.5681	679	26.0576	8.7893	729	27.	9.
630	25.0998	8.5726	680	26.0768	8.7937	730	27.0185	9.0041
631	25.1197	8.5772	681	26.0960	8.7980	731	27.0370	9.0082
632	25.1396	8.5817	682	26.1151	8.8023	732	27.0555	9.0123
633	25.1595	8.5862	683	26.1343	8.8066	733	27.0740	9.0164
634	25.1794	8.5907	684	26.1534	8.8109	734	27.0924	9.0205
635	25.1992	8.5952	685	26.1725	8.8152	735	27.1109	9.0246

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
736	27.1293	9.0287	786	28.0357	9.2287	836	28.9137	9.4204
737	27.1477	9.0328	787	28.0535	9.2326	837	28.9310	9.4241
738	27.1662	9.0369	788	28.0713	9.2365	838	28.9482	9.4279
739	27.1846	9.0410	789	28.0891	9.2404	839	28.9655	9.4316
740	27.2029	9.0450	790	28.1069	9.2443	840	28.9828	9.4354
741	27.2213	9.0491	791	28.1247	9.2482	841	29.	9.4391
742	27.2397	9.0532	792	28.1425	9.2521	842	29.0172	9.4429
743	27.2580	9.0572	793	28.1603	9.2560	843	29.0345	9.4466
744	27.2764	9.0613	794	28.1780	9.2599	844	29.0517	9.4503
745	27.2947	9.0654	795	28.1957	9.2638	845	29.0689	9.4541
746	27.3130	9.0694	796	28.2135	9.2677	846	29.0861	9.4578
747	27.3313	9.0735	797	28.2312	9.2716	847	29.1033	9.4615
748	27.3496	9.0775	798	28.2489	9.2754	848	29.1204	9.4652
749	27.3679	9.0816	799	28.2666	9.2793	849	29.1376	9.4690
750	27.3861	9.0856	800	28.2843	9.2832	850	29.1548	9.4727
751	27.4044	9.0896	801	28.3019	9.2870	851	29.1719	9.4764
752	27.4226	9.0937	802	28.3196	9.2909	852	29.1890	9.4801
753	27.4408	9.0977	803	28.3373	9.2948	853	29.2062	9.4838
754	27.4591	9.1017	804	28.3549	9.2986	854	29.2233	9.4875
755	27.4773	9.1057	805	28.3725	9.3025	855	29.2404	9.4912
756	27.4955	9.1098	806	28.3901	9.3036	856	29.2575	9.4949
757	27.5136	9.1138	807	28.4077	9.3102	857	29.2746	9.4986
758	27.5318	9.1178	808	28.4253	9.3140	858	29.2916	9.5023
759	27.5500	9.1218	809	28.4429	9.3179	859	29.3087	9.5060
760	27.5681	9.1258	810	28.4605	9.3217	860	29.3258	9.5097
761	27.5862	9.1298	811	28.4781	9.3255	861	29.3428	9.5134
762	27.6043	9.1338	812	28.4956	9.3294	862	29.3589	9.5171
763	27.6225	9.1378	813	28.5132	9.3332	863	29.3769	9.5207
764	27.6405	9.1418	814	28.5307	9.3370	864	29.3939	9.5244
765	27.6586	9.1458	815	28.5482	9.3408	865	29.4109	9.5281
766	27.6767	9.1498	816	28.5657	9.3447	866	29.4279	9.5317
767	27.6948	9.1537	817	28.5832	9.3485	867	29.4449	9.5354
768	27.7128	9.1577	818	28.6007	9.3523	868	29.4618	9.5391
769	27.7308	9.1617	819	28.6182	9.3561	869	29.4788	9.5427
770	27.7489	9.1657	820	28.6356	9.3599	870	29.4958	9.5464
771	27.7669	9.1696	821	28.6531	9.3637	871	29.5127	9.5501
772	27.7849	9.1736	822	28.6705	9.3675	872	29.5296	9.5537
773	27.8029	9.1775	823	28.6880	9.3713	873	29.5466	9.5574
774	27.8209	9.1815	824	28.7051	9.3751	874	29.5635	9.5610
775	27.8388	9.1855	825	28.7288	9.3789	875	29.5804	9.5647
776	27.8568	9.1894	826	28.7402	9.3827	876	29.5973	9.5683
777	27.8747	9.1933	827	28.7576	9.3865	877	29.6142	9.5719
778	27.8927	9.1973	828	28.7750	9.3902	878	29.6311	9.5756
779	27.9106	9.2012	829	28.7924	9.3940	879	29.6479	9.5792
780	27.9285	9.2052	830	28.8097	9.3978	880	29.6648	9.5828
781	27.9464	9.2091	831	28.8271	9.4016	881	29.6816	9.5865
782	27.9643	9.2130	832	28.8444	9.4053	882	29.6985	9.5901
783	27.9821	9.2170	833	28.8617	9.4091	883	29.7153	9.5937
784	28.	9.2209	834	28.8791	9.4129	884	29.7321	9.5973
785	28.0179	9.2248	835	28.8964	9.4166	885	29.7489	9.6010

TABLE OF SQUARE ROOTS AND CUBE ROOTS.—Continued.

No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.	No.	Sq. Rt.	Cu. Rt.
886	29.7658	9.6046	926	30.4302	9.7470	966	31.0805	9.8854
887	29.7825	9.6082	927	30.4467	9.7505	967	31.0966	9.8888
888	29.7993	9.6118	928	30.4631	9.7540	968	31.1127	9.8922
889	29.8161	9.6154	929	30.4795	9.7575	969	31.1288	9.8956
890	29.8329	9.6190	930	30.4959	9.7610	970	31.1448	9.8990
891	29.8496	9.6226	931	30.5123	9.7645	971	31.1609	9.9024
892	29.8664	9.6262	932	30.5287	9.7680	972	31.1769	9.9058
893	29.8831	9.6298	933	30.5450	9.7715	973	31.1929	9.9092
894	29.8998	9.6334	934	30.5614	9.7750	974	31.2090	9.9126
895	29.9166	9.6370	935	30.5778	9.7785	975	31.2250	9.9160
896	29.9333	9.6406	936	30.5941	9.7829	976	31.2410	9.9194
897	29.9500	9.6442	937	30.6105	9.7854	977	31.2570	9.9227
898	29.9666	9.6477	938	30.6268	9.7889	978	31.2730	9.9261
899	29.9833	9.6513	939	30.6431	9.7924	979	31.2890	9.9295
900	30.	9.6549	940	30.6594	9.7959	980	31.3050	9.9329
901	30.0167	9.6585	941	30.6757	9.7993	981	31.3209	9.9363
902	30.0333	9.6620	942	30.6920	9.8028	982	31.3369	9.9396
903	30.0500	9.6656	943	30.7083	9.8063	983	31.3528	9.9430
904	30.0666	9.6692	944	30.7246	9.8097	984	31.3688	9.9464
905	30.0832	9.6727	945	30.7409	9.8132	985	31.3847	9.9497
906	30.0998	9.6763	946	30.7571	9.8167	986	31.4006	9.9531
907	30.1164	9.6799	947	30.7734	9.8201	987	31.4166	9.9565
908	30.1330	9.6834	948	30.7896	9.8236	988	31.4325	9.9589
909	30.1496	9.6870	949	30.8058	9.8270	989	31.4484	9.9632
910	30.1662	9.6905	950	30.8221	9.8305	990	31.4643	9.9666
911	30.1828	9.6941	951	30.8383	9.8339	991	31.4802	9.9699
912	30.1993	9.6976	952	30.8545	9.8374	992	31.4960	9.9733
913	30.2159	9.7012	953	30.8707	9.8408	993	31.5119	9.9766
914	30.2324	9.7047	954	30.8869	9.8443	994	31.5278	9.9800
915	30.2490	9.7082	955	30.9031	9.8477	995	31.5436	9.9833
916	30.2655	9.7118	956	30.9192	9.8511	996	31.5595	9.9866
917	30.2820	9.7153	957	30.9354	9.8546	997	31.5753	9.9900
918	30.2985	9.7188	958	30.9516	9.8580	998	31.5911	9.9933
919	30.3150	9.7224	959	30.9677	9.8614	999	31.6070	9.9967
920	30.3315	9.7259	960	30.9839	9.8648	1000	31.6228	10.
921	30.3480	9.7294	961	31.	9.8683			
922	30.3645	9.7329	962	31.0161	9.8717			
923	30.3809	9.7364	963	31.0322	9.8751			
924	30.3974	9.7400	964	31.0483	9.8785			
925	30.4138	9.7435	965	31.0644	9.8819			



Radius	$= A - B = A - D = A\ F$
Sine	$= C - D = A - E$
Tangent	$= B - G$
Versed sine	$= B - C$
Secant	$= A - G$
Co-sine	$= A - C = D - E$
Co-tangent	$= F - H$
Co-versine	$= E - F$
Co-secant	$= A - H$

TRIGONOMETRICAL FORMULAE.

Sine $= \sqrt{1 - \text{co-sine}^2} = \dfrac{1}{\text{co-secant}}$.

Co-sine $= \sqrt{1 - \text{sine}^2} = \dfrac{1}{\text{secant.}}$

Tangent $= \dfrac{\text{sine}}{\text{co-sine}} = \dfrac{1}{\text{co-tangent.}}$

Co-tangent $= \dfrac{\text{co-sine}}{\text{sine}} = \dfrac{1}{\text{tangent.}}$

Secant $= \sqrt{\text{radius}^2 + \text{tangent}^2} = \dfrac{\quad}{\text{co-sine.}}$

Co-secant $= \dfrac{1}{\text{sine.}}$

Versed sine = radius — co-sine.

Co-versed sine = radius — sine.

Radius $= \sqrt{\text{sine}^2 + \text{co-sine}^2}$.

NATURAL SINES.

DEGREES.

Min'ts	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	Min'ts.
0	.00000	.01745	.03490	.05234	.06976	.08716	.10453	.12187	.13917	.15643	.17365	.19081	.20791	.22495	.24192	60
5	.00145	.01891	.03635	.05379	.07121	.08860	.10597	.12331	.14061	.15787	.17508	.19224	.20933	.22637	.24333	55
10	.00291	.02036	.03780	.05524	.07266	.09005	.10742	.12476	.14205	.15931	.17651	.19366	.21076	.22778	.24474	50
15	.00436	.02181	.03926	.05669	.07411	.09150	.10887	.12620	.14349	.16074	.17794	.19509	.21218	.22920	.24615	45
20	.00582	.02327	.04071	.05814	.07556	.09295	.11031	.12764	.14493	.16218	.17937	.19652	.21360	.23062	.24756	40
25	.00727	.02472	.04217	.05960	.07701	.09440	.11176	.12908	.14637	.16361	.18080	.19794	.21502	.23203	.24897	35
30	.00873	.02618	.04362	.06105	.07846	.09585	.11320	.13053	.14781	.16504	.18224	.19937	.21644	.23345	.25038	30
35	.01018	.02763	.04507	.06250	.07991	.09729	.11465	.13197	.14925	.16648	.18367	.20079	.21786	.23486	.25179	25
40	.01163	.02908	.04652	.06395	.08136	.09874	.11609	.13341	.15069	.16792	.18509	.20222	.21928	.23627	.25319	20
45	.01309	.03054	.04798	.06540	.08281	.10019	.11754	.13485	.15212	.16935	.18652	.20364	.22070	.23769	.25460	15
50	.01454	.03199	.04943	.06685	.08426	.10163	.11898	.13629	.15356	.17078	.18795	.20507	.22212	.23910	.25601	10
55	.01600	.03345	.05088	.06830	.08571	.10308	.12043	.13773	.15500	.17222	.18938	.20649	.22353	.24051	.25741	5
60	.01745	.03490	.05234	.06976	.08716	.10453	.12187	.13917	.15643	.17365	.19081	.20791	.22495	.24192	.25882	0
	89	88	87	86	85	84	83	82	81	80	79	78	77	76	75	

DEGREES.

NATURAL CO-SINES.

NATURAL SINES.

DEGREES.

Min'ts.	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	Min'ts.
0	.25882	.27564	.29237	.30902	.32557	.34202	.35837	.37461	.39073	.40674	.42262	.43837	.45399	.46947	.48481	60
5	.26022	.27703	.29376	.31040	.32694	.34339	.35972	.37595	.39207	.40806	.42394	.43968	.45529	.47075	.48608	55
10	.26163	.27843	.29515	.31178	.32832	.34475	.36108	.37730	.39341	.40939	.42525	.44098	.45658	.47204	.48735	50
15	.26303	.27983	.29654	.31316	.32969	.34612	.36244	.37865	.39474	.41072	.42657	.44229	.45787	.47332	.48862	45
20	.26443	.28122	.29793	.31454	.33106	.34748	.36379	.37999	.39608	.41204	.42788	.44359	.45917	.47460	.48989	40
25	.26584	.28262	.29932	.31592	.33244	.34884	.36515	.38134	.39741	.41337	.42920	.44490	.46046	.47588	.49116	35
30	.26724	.28401	.30071	.31730	.33381	.35021	.36650	.38268	.39875	.41469	.43051	.44620	.46175	.47716	.49242	30
35	.26864	.28541	.30209	.31868	.33518	.35157	.36785	.38403	.40008	.41602	.43182	.44750	.46304	.47844	.49369	25
40	.27004	.28680	.30348	.32006	.33655	.35293	.36921	.28537	.40141	.41734	.43313	.44880	.46433	.47971	.49495	20
45	.27144	.28820	.30486	.32144	.33792	.35429	.37056	.38671	.40275	.41866	.43444	.45010	.46561	.48099	.49622	15
50	.27284	.28959	.30625	.32282	.23928	.35565	.37191	.38805	.40408	.41998	.43575	.45140	.46690	.48226	.49748	10
55	.27424	.29098	.30763	.32419	.34065	.35701	.37326	.38939	.40541	.42130	.43706	.45269	.46819	.48354	.49874	5
60	.27564	.29237	.30902	.32557	.34202	.35837	.37461	.39073	.40674	.42262	.43837	.45399	.46947	.48481	.50000	0
	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60	

DEGREES.

NATURAL CO-SINES.

NATURAL SINES.

DEGREES.

Min'ts.	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	Min'ts.
0	.50000	.51504	.52992	.54464	.55919	.57358	.58778	.60181	.61566	.62932	.64279	.65606	.66913	.68200	.69466	60
5	.50126	.51628	.53115	.54586	.56040	.57477	.58896	.60298	.61681	.63045	.64390	.65716	.67021	.68306	.69570	55
10	.50252	.51753	.53238	.54708	.56160	.57596	.59014	.60414	.61795	.63158	.64501	.65825	.67129	.68412	.69675	50
15	.50377	.51877	.53361	.54829	.56280	.57714	.59131	.60529	.61909	.63270	.64612	.65935	.67237	.68518	.69779	45
20	.50503	.52002	.53484	.54951	.56401	.57833	.59248	.60645	.62024	.63383	.64723	.66044	.67344	.68624	.69883	40
25	.50628	.52126	.53607	.55072	.56521	.57952	.59365	.60761	62138	.63495	.64834	.66153	.67452	.68730	.69987	35
30	.50754	.52250	.53730	.55194	.56641	.58070	.59482	.60876	.62251	.63608	.64945	.66262	.67559	.68835	.70091	30
35	.50879	.52374	.53853	.55315	.56760	.58189	.59599	.60991	.52365	.63720	.65055	.66371	67666	.68941	.70195	25
40	.51004	.52498	.53975	.55436	.56880	.58307	.59716	.61107	.62479	.63832	.65166	.66480	.67773	.69046	.70298	20
45	.51129	.52621	.54097	.55557	.57000	.58425	.59832	.61222	.62592	.63944	.65276	.66588	.67880	.69151	.70401	15
50	.51254	.52745	.54220	.55678	.57119	.58543	.59949	.61337	.62706	.64056	.65386	.66697	.67987	.69256	.70505	10
55	.51379	.52868	.54342	.55799	.57238	.58661	.60065	.61451	.62819	.64167	.65496	.66805	.68093	.69361	.70608	5
60	.51504	.52992	.54464	.55919	.57358	.58778	.60181	.61566	.62932	.64279	.65606	.66913	.68200	.69466	.70701	0
	59	58	57	56	55	54	53	52	51	50	49	48	47	46	45	

DEGREES.

NATURAL CO-SINES.

NATURAL SINES.

DEGREES.

Min'ts.	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	Min'ts.
0	.70711	.71934	.73135	.74314	.75471	.76601	.77715	.78801	.79864	.80902	.81915	.82904	.83867	.84805	.85717	60
5	.70813	.72035	.73234	.74412	.75566	.76698	.77806	.78891	.79951	.80987	.81999	.82985	.83946	.84882	.85792	55
10	.70916	.72136	.73333	.74509	.75661	.76791	.77897	.78980	.80038	.81072	.82082	.83066	.84025	.84959	.85866	50
15	.71019	.72236	.73432	.74605	.75756	.76884	.77988	.79069	.80125	.81157	.82165	.83147	.84104	.85035	.85941	45
20	.71121	.72337	.73531	.74702	.75851	.76977	.78079	.79158	.80212	.81242	.82247	.83228	.84182	.85112	.86015	40
25	.71223	.72437	.73629	.74799	.75946	.77070	.78170	.79247	.80299	.81327	.82330	.83308	.84261	.85188	.86089	35
30	.71325	.72537	.73728	.74896	.76041	.77162	.78261	.79335	.80386	.81412	.82413	.83389	.84339	.85264	.86163	30
35	.71427	.72637	.73826	.74992	.76135	.77255	.78351	.79424	.80472	.81496	.82495	.83469	.84417	.85340	.86237	25
40	.71529	.72737	.73924	.75088	.76229	.77347	.78442	.79512	.80558	.81580	.82577	.83549	.84495	.85416	.86310	20
45	.71630	.72837	.74022	.75184	.76323	.77439	.78532	.79600	.80644	.81664	.82659	.83629	.84573	.85491	.86384	15
50	.71732	.72937	.74120	.75280	.76417	.77531	.78622	.79688	.80730	.81748	.82741	.83708	.84650	.85567	.86457	10
55	.71833	.73036	.74217	.75375	.76511	.77623	.78711	.79776	.80816	.81832	.82822	.83788	.84728	.85642	.86530	5
60	.71934	.73135	.74314	.75471	.76604	.77715	.78801	.79864	.80902	.81915	.82904	.83867	.84805	.85717	.86603	0
	44	43	42	41	40	39	38	37	36	35	34	33	32	31	30	

DEGREES.

NATURAL CO-SINES.

NATURAL SINES.

DEGREES.

Min'ts.	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	Min'ts.
0	.86603	.87462	.88295	.89101	.89879	.90631	.91355	.92050	.92718	.93358	.93969	.94552	.95106	.95630	.96126	60
5	.86675	.87532	.88363	.89167	.89943	.90692	.91414	.92107	.92773	.9341	.94019	.94599	.95150	.95673	.96166	55
10	.86748	.87603	.88431	.89232	.90006	.90753	.91472	.92164	.92827	.93462	.94068	.94646	.95195	.95715	.96206	50
15	.86820	.87673	.88499	.89298	.90070	.90814	.91531	.92220	.92881	.93514	.94118	.94693	.95240	.95757	.96246	45
20	.86892	.87743	.88566	.89363	.90133	.90875	.91590	.92276	.92935	.93565	.94167	.94740	.95284	.95799	.96285	40
25	.86964	.87812	.88634	.89428	.90195	.90936	.91648	.92332	.92988	.93616	.94215	.94786	.95328	.95841	.96324	35
30	.87036	.87882	.88701	.89493	.90259	.90996	.91706	.92388	.93042	.93667	.94264	.94832	.95372	.95882	.96363	30
35	.87107	.87951	.88768	.89558	.90321	.91056	.91764	.92443	.93095	.93718	.94313	.94878	.95415	.95923	.96402	25
40	.87178	.88020	.88835	.89623	.90383	.91116	.91822	.92499	.93148	.93769	.94361	.94924	.95459	.95964	.96440	20
45	.87250	.88089	.88902	.89687	.90445	.91176	.91879	.92554	.93201	.93819	.94409	.94970	.95502	.96005	.96479	15
50	.87321	.88158	.88968	.89751	.90507	.91236	.91936	.92609	.93253	.93869	.94457	.95015	.95545	.96046	.96517	10
55	.87391	.88226	.89035	.89815	.90569	.91295	.91994	.92664	.93306	.93919	.94504	.95061	.95588	.96086	.96555	5
60	.87462	.88295	.89101	.89879	.90631	.91355	.92050	.92718	.93358	.93969	.94552	.95106	.95630	.96126	.96593	0
	29	28	27	26	25	24	23	22	21	20	19	18	17	16	15	

DEGREES.

NATURAL CO-SINES.

NATURAL SINES.

DEGREES.

Min'ts	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	Min'ts.
0	.96593	.97030	.97437	.97815	.98163	.98481	.98769	.99027	.99255	.99452	.99619	.99756	.99863	.99939	.99985	60
5	.96630	.97065	.97470	.97845	.98190	.98506	.98791	.99047	.99272	.99467	.99632	.99766	.99870	.99944	.99987	55
10	.96667	.97100	.97502	.97875	.98218	.98531	.98814	.99067	.99290	.99482	.99644	.99776	.99878	.99949	.99989	50
15	.96705	.97134	.97534	.97905	.98245	.98556	.98836	.99087	.99307	.99497	.99657	.99786	.99885	.99953	.99991	45
20	.96741	.97169	.97566	.97934	.98272	.98580	.98858	.99106	.99324	.99511	.99668	.99795	.99892	.99958	.99993	40
25	.96778	.97203	.97598	.97963	.98299	.98604	.98880	.99125	.99341	.99526	.99680	.99804	.99898	.99962	.99995	35
30	.96815	.97237	.97630	.97992	.98325	.98629	.98902	.99144	.99357	.99540	.99692	.99813	.99905	.99966	.99996	30
35	.96851	.97271	.97661	.98021	.98352	.98652	.98923	.99163	.99374	.99553	.99703	.99822	.99911	.99969	.99997	25
40	.96887	.97304	.97692	.98050	.98378	.98676	.98944	.99182	.99390	.99567	.99714	.99831	.99917	.99973	.99998	20
45	.96923	.97338	.97723	.98079	.98404	.98700	.98965	.99200	.99406	.99580	.99725	.99839	.99923	.99976	.99999	15
50	.96959	.97371	.97754	.98107	.98430	.98723	.98986	.99219	.99421	.99594	.99736	.99847	.99928	.99979	.99999	10
55	.96994	.97404	.97784	.98135	.98455	.98746	.99006	.99237	.99437	.99607	.99746	.99855	.99934	.99982	.99999	5
60	.97030	.97437	.97815	.98163	.98481	.98769	.99027	.99255	.99452	.99619	.99756	.99863	.99939	.99985	1.00000	0
	14	13	12	11	10	9	8	7	6	5	4	3	2	1	0	

DEGREES.

NATURAL CO-SINES.

NATURAL TANGENTS.

DEGREES.

Min'ts.	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	Min'ts.
0	.00000	.01745	.03492	.05241	.06993	.08749	.10510	.12278	.14054	.15838	.17633	.19438	.21256	.23087	.24933	60
5	.00145	.01891	.03638	.05387	.07139	.08895	.10657	.12426	.14202	.15987	.17783	.19589	.21408	.23240	.25087	55
10	.00291	.02036	.03783	.05532	.07285	.09042	.10805	.12574	.14351	.16137	.17933	.19740	.21560	.23393	.25242	50
15	.00436	.02182	.03929	.05678	.07431	.09189	.10952	.12722	.14499	.16286	.18083	.19891	.21712	.23547	.25397	45
20	.00582	.02327	.04075	.05824	.07577	.09335	.11099	.12869	.14648	.16435	.18233	.20042	.21864	.23700	.25552	40
25	.00727	.02473	.04220	.05970	.07724	.09482	.11246	.13017	.14796	.16585	.18383	.20194	.22017	.23854	.25707	35
30	.00873	.02618	.04366	.06116	.07870	.09629	.11393	.13165	.14945	.16734	.18534	.20345	.22169	.24208	.25862	30
35	.01018	.02764	.04512	.06262	.08016	.09776	.11541	.13313	.15094	.16884	.18684	.20497	.22322	.24162	.26017	25
40	.01164	.02910	.04657	.06408	.08163	.09922	.11688	.13461	.15243	.17033	.18835	.20648	.22475	.24316	.26172	20
45	.01309	.03055	.04803	.06554	.08309	.10069	.11836	.13609	.15391	.17183	.18985	.20800	.22628	.24470	.26328	15
50	.01454	.03201	.04949	.06700	.08456	.10216	.11983	.13757	.15540	.17333	.19136	.20952	.22781	.24624	.26483	10
55	.01600	.03346	.05095	.06846	.08602	.10363	.12131	.13906	.15689	.17483	.19287	.21104	.22934	.24778	.26639	5
60	.01745	.03492	.05241	.06993	.08749	.10510	.12278	.14054	.15838	.17633	.19438	.21256	.23087	.24933	.26795	0
	89	88	87	86	85	84	83	82	81	80	79	78	77	76	75	

DEGREES.

NATURAL CO-TANGENTS.

NATURAL TANGENTS.

DEGREES.

Min'ts.	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	Min'ts.
0	.26795	.28674	.30573	.32492	.34433	.36397	.38386	.40403	.42447	.44523	.46631	.48773	.50952	.53171	.55431	60
5	.26951	.28832	.30732	.32653	.34595	.36562	.38553	.40572	.42619	.44697	.46808	.48953	.51136	.53358	.55621	55
10	.27107	.28990	.30891	.32814	.34758	.36727	.38720	.40741	.42791	.44872	.46985	.49134	.51319	.53545	.55812	50
15	.27263	.29147	.31051	.32975	.34921	.36892	.38888	.40911	.42963	.45047	.47163	.49314	.51503	.53732	.56003	45
20	.27419	.29305	.31210	.33136	.35085	.37057	.39055	.41081	.43136	.45222	.47341	.49495	.51687	.53919	.56194	40
25	.27576	.29463	.31370	.33298	.35248	.37223	.39223	.41251	.43308	.45397	.47519	.49677	.51872	.54107	.56385	35
30	.27732	.29621	.31530	.33459	.35412	.37388	.39391	.41421	.43481	.45573	.47697	.49858	.52057	.54295	.56577	30
35	.27889	.29780	.31690	.33621	.35576	.37554	.39559	.41592	.43654	.45748	.47876	.50040	.52242	.54484	.56769	25
40	.28046	.29938	.31850	.33783	.35739	.37720	.39727	.41762	.43827	.45924	.48055	.50222	.52427	.54673	.56962	20
45	.28203	.30096	.32010	.33945	.35904	.37887	.39896	.41933	.44001	.46101	.48234	.50404	.52612	.54862	.57155	15
50	.28360	.30255	.32171	.34108	.36068	.38053	.40065	.42105	.44175	.46277	.48414	.50587	.52798	.55051	.57348	10
55	.28517	.30446	.32331	.34270	.36232	.28220	.40233	.42276	.44349	.40454	.48593	.50769	.52984	.55241	.57541	5
60	.28674	.30573	.32492	.34433	.36397	.38386	.40403	.42447	.44523	.46631	.48773	.50952	.53171	.55431	.57735	0
	74	73	72	71	70	69	68	67	66	65	64	63	62	61	60	

DEGREES.

NATURAL CO-TANGENTS.

NATURAL TANGENTS.

DEGREES.

Min'ts.	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	Min'ts.
0	.57735	.60086	.62487	.64941	.67451	.70021	.72654	.75355	.78128	.80978	.83910	.86929	.90040	.93251	.96569	60
5	.57929	.60284	.62689	.65148	.67663	.70238	.72877	.75584	.78363	.81219	.84158	.87184	.90304	.93524	.96850	55
10	.58123	.60483	.62892	.65355	.67875	.70455	.73100	.75812	.78598	.81461	.84407	.87441	.90568	.93797	.97133	50
15	.58318	.60681	.63095	.65563	.68087	.70673	.73323	.76042	.78834	.81703	.84656	.87698	.90834	.94071	.97416	45
20	.58513	.60881	.63299	.65771	.68301	.70891	.73547	.76271	.79070	.81946	.84906	.87955	.91099	.94345	.97699	40
25	.58709	.61080	.63503	.65980	.68514	.71110	.73771	.76502	.79306	.82190	.85157	.88213	.91366	.94620	.97984	35
30	.58904	.61280	.63707	.66188	.68728	.71329	.73996	.76733	.79543	.82434	.85408	.88472	.91633	.94896	.98270	30
35	.59100	.61480	.63912	.66398	.68942	.71549	.74221	.76964	.79781	.82678	.85660	.88732	.91901	.95173	.98556	25
40	.59297	.61681	.64117	.66608	.69157	.71769	.74447	.77196	.80020	.82923	.85912	.88992	.92170	.95451	.98843	20
45	.59494	.61882	.64322	.66818	.69372	.71990	.74673	.77428	.80258	.83169	.86165	.89253	.92439	.95729	.99131	15
50	.59691	.62083	.64528	.67028	.69588	.72211	.74900	.77661	.80498	.83415	.86419	.89515	.92709	.96008	.99420	10
55	.59888	.62285	.64734	.67239	.69804	.72432	.75128	.77895	.80738	.83662	.86674	.89777	.92980	.96288	.99709	5
60	.60086	.62487	.64941	.67451	.70021	.72654	.75355	.78128	.80978	.83910	.86929	.90040	.93251	.96569	1.00000	0
	59	58	57	56	55	54	53	52	51	50	49	48	47	46	45	

DEGREES.

NATURAL CO-TANGENTS.

NATURAL TANGENTS.

DEGREES.

Min's	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	Min's
0	1.00000	1.03553	1.07237	1.11061	1.15037	1.19175	1.23490	1.27994	1.32704	1.37638	1.42815	1.48256	1.53986	1.60033	1.66428	60
5	1.00291	1.03855	1.07550	1.11387	1.15375	1.19528	1.23858	1.28379	1.33107	1.38060	1.43258	1.48722	1.54478	1.60553	1.66977	55
10	1.00583	1.04158	1.07864	1.11713	1.15715	1.19882	1.24227	1.28764	1.33511	1.38483	1.43703	1.49190	1.54971	1.61074	1.67530	50
15	1.00876	1.04461	1.08179	1.12040	1.16056	1.20237	1.24597	1.29152	1.33916	1.38909	1.44149	1.49660	1.55467	1.61598	1.68085	45
20	1.01170	1.04766	1.08495	1.12369	1.16398	1.20593	1.24969	1.29540	1.34323	1.39336	1.44598	1.50133	1.55965	1.62125	1.68643	40
25	1.01465	1.05071	1.08813	1.12699	1.16741	1.20951	1.25343	1.29931	1.34732	1.39764	1.45048	1.50607	1.56466	1.62654	1.69203	35
30	1.01761	1.05378	1.09131	1.13029	1.17085	1.21310	1.25717	1.30322	1.35142	1.40195	1.45501	1.51083	1.56968	1.63185	1.69766	30
35	1.02057	1.05685	1.09450	1.13361	1.17430	1.21670	1.26093	1.30716	1.35554	1.40627	1.45955	1.51562	1.57473	1.63719	1.70332	25
40	1.02354	1.05994	1.09770	1.13694	1.17777	1.22031	1.26471	1.31110	1.35968	1.41061	1.46411	1.52043	1.57981	1.64256	1.70901	20
45	1.02653	1.06303	1.10091	1.14028	1.18125	1.22394	1.26849	1.31507	1.36383	1.41497	1.46870	1.52525	1.58490	1.64795	1.71473	15
50	1.02952	1.06613	1.10414	1.14363	1.18474	1.22758	1.27229	1.31904	1.36799	1.41934	1.47330	1.53010	1.59002	1.65337	1.72047	10
55	1.03252	1.06925	1.10737	1.14699	1.18824	1.23123	1.27611	1.32304	1.37218	1.42374	1.47792	1.53497	1.59517	1.65881	1.72625	5
60	1.03553	1.07237	1.11061	1.15037	1.19175	1.23490	1.27994	1.32704	1.37638	1.42815	1.48256	1.53986	1.60033	1.66428	1.73205	0
	44	43	42	41	40	39	38	37	36	35	34	33	32	31	30	

DEGREES.

NATURAL CO-TANGENTS.

NATURAL TANGENTS.

DEGREES.

Mins.	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	Mins.
0	1.73205	1.80405	1.88073	1.96261	2.05030	2.14451	2.24604	2.35585	2.47509	2.60509	2.74748	2.90421	3.07768	3.27085	3.48741	60
5	1.73788	1.81025	1.88734	1.96969	2.05789	2.15267	2.25486	2.36541	2.48549	2.61646	2.75996	2.91799	3.09298	3.28795	3.50665	55
10	1.74374	1.81649	1.89400	1.97680	2.06553	2.16089	2.26373	2.37504	2.49597	2.62791	2.77254	2.93189	3.10842	3.30521	3.52609	50
15	1.74964	1.82276	1.90069	1.98396	2.07321	2.16917	2.27267	2.38473	2.50652	2.63945	2.78523	2.94590	3.12400	3.32264	3.54573	45
20	1.75556	1.82906	1.90741	1.99116	2.08094	2.17749	2.28167	2.39449	2.51715	2.65109	2.79802	2.96004	3.13972	3.34023	3.56557	40
25	1.76151	1.83540	1.91418	1.99840	2.08872	2.18587	2.29072	2.40432	2.52786	2.66281	2.81091	2.97430	3.15558	3.35800	3.58562	35
30	1.76749	1.84177	1.92098	2.00569	2.09654	2.19430	2.29984	2.41421	2.53865	2.67462	2.82391	2.98868	3.17159	3.37594	3.60588	30
35	1.77351	1.84818	1.92782	2.01302	2.10441	2.20278	2.30902	2.42418	2.54952	2.68653	2.83702	3.00319	3.18775	3.39406	3.62636	25
40	1.77955	1.85461	1.93470	2.02039	2.11233	2.21132	2.31826	2.43422	2.56046	2.69852	2.85023	3.30173	3.20406	3.41236	3.64705	20
45	1.78563	1.86109	1.94162	2.02780	2.12030	2.21992	2.32756	2.44432	2.57149	2.71062	2.86356	3.03259	3.22053	3.43084	3.66796	15
50	1.79174	1.86760	1.94858	2.03526	2.12832	2.22857	2.33693	2.45451	2.58261	2.72281	2.87700	3.04749	3.23714	3.44951	3.68909	10
55	1.79787	1.87414	1.95557	2.04276	2.13639	2.23727	2.34636	2.46476	2.59381	2.73509	2.89055	3.06252	3.25392	3.46837	3.71045	5
60	1.80405	1.88073	1.96261	2.05030	2.14451	2.24604	2.35585	2.47509	2.60509	2.74748	2.90421	3.07768	3.27085	3.48741	3.73205	0
	29	28	27	26	25	24	23	22	21	20	19	18	17	16	15	

DEGREES.

NATURAL CO-TANGENTS.

NATURAL TANGENTS.

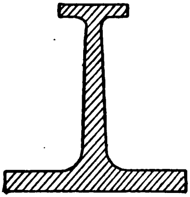
DEGREES.

Min's	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	Min's
0	3.73205	4.01078	4.33147	4.70463	5.14455	5.67128	6.31375	7.11537	8.14435	9.51436	11.43005	14.30066	19.08113	28.63625	57.28996	60
5	3.75388	4.03578	4.36040	4.73851	5.18480	5.71992	6.37373	7.19124	8.24345	9.64935	11.62476	14.60591	19.62729	29.88229	62.49915	55
10	3.77595	4.06107	4.38969	4.77286	5.22566	5.76937	6.43484	7.26872	8.34495	9.78817	11.82616	14.92441	20.20555	31.24157	68.75008	50
15	3.79827	4.08666	4.41936	4.80768	5.26715	5.81966	6.49710	7.34786	8.44896	9.93101	12.03462	15.25705	20.81882	32.73026	76.39000	45
20	3.82083	4.11256	4.44942	4.84300	5.30928	5.87080	6.56055	7.42871	8.55555	10.07803	12.25050	15.60478	21.47040	34.36777	85.93979	40
25	3.84364	4.13877	4.47986	4.87882	5.35206	5.92283	6.62522	7.51132	8.66482	10.22942	12.47422	15.96866	22.16398	36.17759	98.21794	35
30	3.86671	4.16530	4.51071	4.91516	5.39552	5.97576	6.69116	7.59575	8.77689	10.38539	12.70620	16.34985	22.90376	38.18845	114.5886	30
35	3.89004	4.19215	4.54196	4.95201	5.43966	6.02962	6.75898	7.68208	8.89185	10.54615	12.94692	16.74961	23.69453	40.43583	137.5075	25
40	3.91364	4.21933	4.57363	4.98940	5.48450	6.08444	6.82694	7.77035	9.00983	10.71191	13.19688	17.16933	24.54175	42.96407	171.8854	20
45	3.93751	4.24685	4.60572	5.02734	5.53007	6.14023	6.89688	7.86064	9.13093	10.88292	13.45662	17.61055	25.45170	45.82935	229.1816	15
50	3.96165	4.27471	4.63824	5.06583	5.57638	6.19702	6.96823	7.95302	9.25530	11.05943	13.72673	18.07497	26.43160	49.10388	343.7737	10
55	3.98607	4.30291	4.67121	5.10490	5.62344	6.25486	7.04105	8.04756	9.38307	11.24171	14.00785	18.56447	27.48985	52.88211	687.5488	5
60	4.01078	4.33147	4.70463	5.14455	5.67128	6.31375	7.11537	8.14435	9.51436	11.43005	14.30066	19.08113	28.63625	57.28996	Infinite	0
	14	13	12	11	10	9	8	7	6	5	4	3	2	1	0	

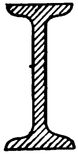
DEGREES.

NATURAL CO-TANGENTS.

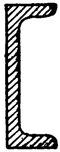
SECTIONS OF IRON BEAMS.



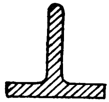
Hodgkinson Cast Iron Beam.



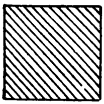
Rolled I Beam.



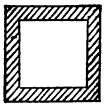
Rolled Channel Beam.



Rolled T Beam.

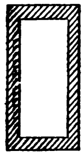


Solid Square Beam.

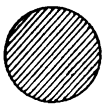


Hollow Square Beam.

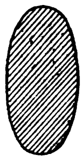
Solid Rectangular Beam.



Hollow Rectangular Beam.



Solid Round Beam.



Solid Elliptical Beam.

HORIZONTAL BEAMS.

Hodgkinson gives a formula for the strength of cast iron beams with solid webs and flanges, as follows:

$$W = \frac{a \times d \times 2.426}{L}$$

Where W = center breaking load in tons of 2000 pounds, a = area in inches of lower flange, d = total depth of beam in inches, and L = clear span or distance between supports in feet.

The above formula, although strictly adapted to what is known as the Hodgkinson beam, is equally applicable to cast iron beams of I section.

In estimating the strength of beams the formula generally employed furnishes a center breaking load. Suppose a given beam, supported at both ends, requires 20 tons as a center breaking load, then twice this, or 40 tons, would be the uniformly distributed breaking load. If the same beam was fixed at *both* ends, then the center breaking load would be 30 tons, and the uniformly distributed breaking load 60 tons, or fifty per cent more than for same beam freely supported.

The same beam firmly fixed at one end and free at the other would require a breaking load at the overhung extremity of 5 tons, or an uniformly distributed load of 10 tons. Whence the relative strength of the several modes of securing beams is:

1. For a beam firmly fixed at both ends, and uniformly loaded.. 150
2. Same beam loaded at center 75
3. For a beam freely supported at both ends, and uniformly loaded.............................. 100
4. Same beam loaded at center 50
5. For a beam firmly fixed at one end, and uniformly loaded.... 25
6. Same beam loaded at overhung end...... .... 12.5

The above values are for same beam differently secured, and the clear overhang of last two beams must be equal to the clear span of first four beams.

Having deduced the value of a beam in tons of center breaking load as for beam 4, then for uniformly distributed load multiply by 2; for beam firmly secured at both ends for center load multiply by 1.5; for same with uniformly distributed load multiply by 3; for beam firmly fixed at one end and loaded at the other multiply by .25; and for same beam uniformly loaded multiply by .5, or by formulae:

For uniform rectangular beam of solid section, freely supported at both ends and loaded at center

$$W = \frac{a \times d \times 1.155\, S}{l}$$

Where S = tensile strength of beam in tons of 2000 pounds per square inch of section.

Same beam with uniformly distributed load

$$W = \frac{a \times d \times 2.31\, S}{l}$$

For uniform rectangular beam of solid section, firmly fixed at both ends and loaded at the center

$$W = \frac{a \times d \times 1.733\, S}{l}$$

Same beam uniformly loaded

$$W = \frac{a \times d \times 3.466\, S}{l}$$

For uniform rectangular beam of solid section, firmly fixed at one end and loaded at the other

$$W = \frac{a \times d \times .28875\, S}{l}$$

And same beam unifomly loaded

$$W = \frac{a \times d \times .5775\, S}{l}$$

For horizontal beams of square section, loaded at center

$$W = \frac{d^3 \times 1.155\, S}{l}$$

W, in all cases representing the breaking load in tons of 2000 pounds; a, the area of section in inches; d, the extreme depth of beam in inches; and l, the clear span in inches.

For beams of cylindrical section estimate the value of a square beam, one side of which equals the diameter of cylindrical beam, and multiply by .68, or by formula:

$$W = \frac{d^3 \times S \times .7854}{l}$$

Suppose a beam of yellow pine 8 inches broad, 11.5 inches deep, and 13 feet 6 inches clear span, what is the center breaking load in tons, estimating S of timber as 3 tons?

$$W = \frac{11.5^2 \times 8 \times (1.155 \times 3)}{162} = 21.395 \text{ tons.}$$

Mr. Trautwine says that a beam of square section, when placed upon edge, or with its diagonal vertical, possesses but .7 the strength of same beam placed upon its side, whilst Mr. D. K. Clark represents by formula the strengths as alike.

"Strength being the first law of architecture," it is always preferable to adopt the coefficients representing the greatest safety.

An elliptical beam possesses .68 of the strength of a rectangular beam, the breadth and depth of which are equal to the short and long diameters of the elliptical section.

Formula for rolled I beams, as adopted by the Phœnix Iron Company for horizontal beams freely supported at both ends, center breaking load in tons:

$$W = \frac{4D \times \left(a + \frac{a'}{6}\right) \times S}{L}$$

Where D = effective depth of beam in feet = separation of the centers of gravity of the two flanges. a = area of one flange in sq. inches, a' = area of stem or web in sq. inches, S = ultimate tensile strength in tons, per sq. inch of section, and L = clear span in feet.

The maximum safe working load per sq. inch of section is taken by the Phœnix Iron Co. at 12,000 pounds, or 6 tons, which with iron of a tensile strength of 60,000 pounds, represents a factor of safety of 5.

DEFLECTION OF BEAMS.

The Phœnix Iron Co. have adopted from Moseley's Mechanics of Engineering and Architecture the following formula for center deflection of rolled I beams:

Beam supported at both ends and uniformly loaded

$$D = \frac{.001\ W'\ L^3}{\left(a + \frac{a'}{6}\right) d^2}$$

Same beam loaded at center

$$D = \frac{.006\ W'\ L^3}{\left(a + \frac{a'}{6}\right) d^2}$$

Where D = deflection in inches at center of beam, W' = load, in pounds, upon beam, L = clear span in feet, a = area in sq. inches of one flange, a' = area in sq. inches of stem or web, and d = separation of centers of gravity of the two flanges in inches.

The deflection of same beam, with one end firmly fixed, and loaded at the other,

$$D' = \frac{.096\ W'\ L^3}{\left(a + \frac{a'}{6}\right) d^2}$$

and uniformly loaded

$$D' = \frac{.036\ W'\ L^3}{\left(a + \frac{a'}{6}\right) d^2}$$

Where D′ = deflection of beam at overhung end.

Mr. D. K. Clark gives the following formulae for the deflection of beams.

For beam of rectangular section loaded at center

$$D = \frac{W\ l^3}{4.62\ b\ d^3\ E}$$

Same beam uniformly loaded

$$D = \frac{W\ l^3}{7.4\ b\ d^3\ E}$$

For beam of cylindrical section of uniform diameter, center load

$$D = \frac{W\ l^3}{3.1416\ d^4\ E}$$

Same beam uniformly loaded

$$D = \frac{.625\ W\ l^3}{3.1416\ d^4\ E}$$

Where D = deflection in inches at center of beam, W = load on beam in tons of 2000 pounds, l = clear span in inches, b = breadth of beam in inches, d = depth of beam in inches, and E = modulus of elasticity in tons of 2000 pounds.

The center deflection of a beam under load, according to the Phoenix Iron Co., should not exceed 1-360 of its length or 1-30 of an inch per foot of clear span, whilst Mr. Trautwine limits the safe deflection to 1-480 of its length or 1-40 of an inch per foot of clear span.

STEEL AND IRON WIRE ROPE.

John A. Roebling's Sons, Trenton, N. J.

Trade Number.	Diameter, in.	Breaking strain, tons of 2,000 pounds.	Circumference of hemp rope of equal strength.	Price per foot, cents.	Breaking strain, tons of 2,000 pounds.	Circumference of hemp rope of equal strength.	Price per foot, cents.
	Iron 7 strands of 19 wires.				Steel 7 str'ds of 19 wires.		
1	2.25	74.	15.5	132.	107.		164
2	2.	65.	14.5	115.	97.		144
3	1.75	54.	13.	100.	78.	15.75	124
4	1.625	43.6	12.	86.	64.	14.5	106
5	1.5	35.	10.75	71.	52.	13.	90
6	1.25	27.2	9.5	58.	39.	12.5	74
7	1.125	20.2	8.	45.	30.	10.	57
8	1.	16.	7.	37.	24.	9.25	46
9	0.875	11.4	6.	31.	20.	8.25	38
10	0.75	8.64	5.	28.	13.	6.5	34
10¼	0.625	5.13	4.5	26.	7.	5.	33
10½	0.5625	4.27	4.	25.	5.	4.25	32
10¾	0.5	3.48	3.75	24.			
	Iron 7 strands 7 wires.				Steel 7 strands 7 wires.		
11	1.5	36.	10.75	60.	50.	13.	74
12	1.375	30.	10.	52.	43.	12	64
13	1.25	25.	9.5	45.	36.	10.75	55
14	1.125	20.	8.25	39.	29.	9.	47
15	1.	16.	7.25	32.	23.	8	40
16	0.875	12.3	6.25	25.	18.	7.5	32
17	0.75	8.8	5.5	20.	13.	6.5	24
18	0.6875	7.6	5	17	11	5.75	20
19	0.625	5.8	4.75	14.	8.5	5.	17
20	0.5	4.1	4.	12.	6.	4.75	15
21	0.4375	2.83	3.25	10.			
22	0.375	2.13	2.75	9.			
23	0.3125	1.65	2.5	8.			
24	0.2812	1.38	2.25	7.			
25	0.25	1.03	2.	6.5			
26	0.2187	0.81	1.75	6			
27	0.1875	0.56	1.5	5.5			

STEEL CABLES FOR SUSPENSION BRIDGES.

Jonh A. Roebling's Sons, Trenton, N. J.

Diameter inches.	Breaking load in tons 2000 pds.	Weight per foot run, pds.
2.625	200	15.
2.5	160	11.
2.375	120	8.5
2.25	107	7.4
2.	96	6.5
1.875	88	6.
1.75	75	5.25
1.625	61	4.25
1.5	50	3.5

SHEAVES AND DRUMS FOR WIRE ROPES.

Least diameter in feet of Sheave or Drum for ropes numbers 1 to 10¾ inclusive.

John A. Roebling's Sons, Trenton N. J.

Trade number.	Sheave, iron rope.	Sheave, steel rope.	Trade number.	Sheave, iron rope.	Sheave, steel rope.
1	8.	9.	8	3.	4.
2	7.	8.	9	2.75	3.75
3	6.5	7.5	10	2.5	3.5
4	5.	6.	10¼	2.	3.
5	4.5	5.5	10½	1.75	2.75
6	4.	5.	10¾	1.5	
7	3.5	4.5			

NOTES ON THE USES OF WIRE ROPE.

JOHN A. ROEBLING'S SONS CO., TRENTON, N. J.

Two kinds of wire rope are manufactured. The most pliable variety contains 19 wires in the strand and is generally used for hoisting and running rope. The ropes with 12 wires and 7 wires in the strand are stiffer, and are better adapted for standing rope, guys and rigging. Orders should state the use of the rope, and advice will be given. Ropes are made up to 3 inches in diam., both of iron and steel, upon special application.

For safe working load allow one-fifth to one-seventh of the ultimate strength, according to speed, so as to get good wear from the rope. When substituting wire rope for hemp rope, it is good economy to allow for the former the same weight per foot which experience has approved for the latter.

Wire rope is as pliable as new hemp rope of the same strength; the former will therefore run over the same sized sheaves and pulleys as the latter. But the greater the diameter of the sheaves, pulleys or drums, the longer wire rope will last. In the construction of machinery for wire rope it will be found good economy to make the drums and sheaves as large as possible. The minimum size of drum is given in a column in the table.

Experience has demonstrated that the wear increases with the speed. It is therefore better to increase the load than the speed.

Wire rope is manufactured either with a wire or a hemp center. The latter is more pliable than the former and will wear better where there is short bending. Orders should specify what kind of center is wanted.

Wire rope must not be coiled or uncoiled like hemp rope. When mounted on a reel, the latter should be mounted on a spindle or flat turn-table to pay off the rope. When forwarded in a small coil without reel, roll it over the ground like a wheel, and run off the rope in that way. All untwisting or kinking must be avoided.

To preserve wire rope, apply raw linseed oil with a piece of sheepskin, wool inside; or mix the oil with equal parts of Spanish brown or lamp-black.

To preserve wire rope under water or under ground, take mineral or vegetable tar, add 1 bushel of fresh slacked lime to 1 barrel of tar, which will neutralize the acid, and boil it well, then saturate the rope with the hot tar. To give the mixture body, add some sawdust.

In no case should *galvanized rope* be used for running rope. One day's use scrapes off the coating of zinc, and rusting proceeds with twice the rapidity.

The grooves of cast iron pulleys and sheaves should be filled with well seasoned blocks of hard wood set on end, to be renewed when worn out. This end wood will save wear and increase adhesion. The smaller pulleys or rollers which support the ropes on inclined planes should be constructed on the same plan. When large sheaves run with very great velocity, the grooves should be lined with leather, set on end, or with india rubber. This is done in the case of all sheaves used in the *transmission of power* between distant points by means of ropes, which frequently run at the rate of 4,000 feet per minute.

Steel ropes are to a certain extent taking the place of iron ropes, where it is a special object to combine lightness with strength.

But in substituting a steel rope for an iron running rope, the the object in view should be to gain an increased wear from the rope rather than to reduce the size.

STRENGTH OF HEMP ROPES.

The old rope makers' formula for ultimate strength of hemp rope is

$$S = 448\,g^2 \div d^2\,4421$$

where S = ultimate strength in pounds,
g = girth in inches.
d = diameter in inches.

Suppose a rope, 6 inches girth, what is the breaking load, or maximum strength?

$$S = 448 \times 6^2 = 16,128 \text{ pounds.}$$

STRENGTH IN POUNDS FOR FULL SECTION.
WEIGHT IN POUNDS PER FATHOM = 6 FEET.

Diam.	Girth.	Strength	Weight.	Diam.	Girth.	Strength	Weight
.25	.785	,276	0.154	3.00*	9.425	39,789	22.140
.375	1.178	,622	0.346	3.25	10.210	46,700	25.984
.5	1.571	1,105	0.615	3.50	10.995	54,160	30.136
.75	2.356	2,487	1.384	3.75	11.781	62,178	34.594
1.00	3.141	4,421	2.460	4.00	12.566	70,739	39.360
1.25	3.927	6,908	3.844	4.25	13.352	79,860	44.434
1.50	4.712	9,947	5.535	4.50	14.137	89,530	49.815
1.75	5.498	13,540	7.534	4.75	14.922	99,754	55.504
2.00	6.283	17,685	9.840	5.00	15.708	110,539	61.504
2.25	7.068	22,384	12.454	5.25	16.493	121,856	67.804
2.50	7.854	27,635	15.376	5.50	17.279	133,740	74.415
2.75	8.639	33,435	18.604	6.00	18.849	159,156	88.560

The weight per fathom of hemp rope of any diameter may be determined by the formula

$$W = d^2\,2.46.$$

where W = weight in pounds per fathom
d = diameter of rope in inches.

BREAKING WEIGHT OF TARRED HEMP ROPES IN POUNDS UPON ENTIRE SECTION.

HAND MADE ROPES. *D. K. Clark.*

Girth.	Diam.	Common hemp.	Russian hemp.	Girth.	Diam.	Common hemp.	Russian hemp.
3″	.95″	4,973	6,048	6″	1.91″	18,144	21,616
3½″	1.11″	7,459	8,669	6½″	2.07″	20,518	23,609
4″	1.27″	8,780	10,461	7″	2.24″	22,937	27,462
4½″	1.43″	10,304	12,432	7½″	2.39″	24,967	30,755
5″	1.59″	13,328	15,859	8″	2.54″	26,880	32,032
5½″	1.75″	15,456	18,614				

MACHINE MADE ROPES.

Girth.	Diam.	Cold register.	Warm register.	Girth.	Diam.	Cold register.	Warm register.
3''	.95''	7,392	8,624	6''	1.91''	28,986	33,152
3½''	1.11''	11,220	11,760	6½''	2.07''	34,630	40,544
4''	1.27''	13,104	15,344	7''	2.24''	40,320	47,040
4½''	1.43''	16,329	19,443	7''	2.39''	46,144	53,984
5''	1.59''	20,496	23,990	8''	2.54''	52,483	61,420
6''	1.75''	24,797	29,120				

Trautwine gives the strength of hemp ropes as 6,000 pounds per sq. inch of section, and manilla ropes as 3,000 pounds per sq. inch of section.

TABLE OF STRENGTH OF CHAINS.

Trautwine.

Diam. of rod of which the links are made.	Weight of chain per ft. run.	Breaking strain of the chain.		Diam. of rod of which the links are made.	Weight of chain per ft. run.	Breaking strain of the chain.	
Inches.	Pds.	Pds.	Tons.	Inches.	Pds.	Pds.	Tons.
3-16	0.325	1731	0.865	1	9.26	49280	24.640
¼	0.579	3069	1.534	1⅛	11.7	59226	29.613
5-16	0 904	4794	2.397	1¼	14.5	73114	36.557
⅜	1.30	6922	3.461	1⅜	17.5	88301	44.150
7-16	1.78	9408	4.704	1½	20.8	105280	52.640
½	2.31	12320	6.160	1⅝	24.4	123514	61 757
9-16	2 93	15590	7.795	1¾	28.4	143293	71.646
⅝	3.62	19219	9.609	1⅞	32.6	164505	82.252
11-16	4 38	23274	11 637	2	37.0	187152	93 760
¾	5.21	27687	13 843	2¼	46.9	224448	112.224
13-16	6 11	32301	16.150	2½	57.9	277088	133.534
⅞	7.10	37632	18.811	2¾	70 0	335328	167.664
15-16	8.14	43277	21.638	3	83 3	398944	199 472

DIMENSIONS OF PHŒNIX BEAMS.

(ROLLED IRON.)

Depth.	Weight per yard.	DIMENSIONS—INCHES.			AREA—SQUARE INCHES.		
		Width of Flange.	Average Thickness of Flange.	Thickness of Stem.	a of Flange	a' of Stem.	Sum of $a + \frac{a'}{6}$
15″	200	5 5-16	1.156	.65	6.142	7.715	7.428
15″	150	4¾	.911	.50	4.330	6.340	5.386
12″	170	5½	1.050	.59	5.777	5.446	6.684
12″	125	4¾	.802	.49	3.810	4.880	4.623
10½″	135	5	.875	.50	4.375	4.750	5.166
10½″	105	4½	.745	.44	3.353	3.793	3.986
9″	150	5⅜	1.039	.60	5.586	3.828	6.224
9″	84	4	.700	.40	2.800	2.800	3.261
9″	70	3½	.680	.31	2.381	2.238	2.754
8″	81	4½	.625	.38	2.812	2.476	3.225
8″	65	4	.527	.35	2.109	2.282	2.489
7″	69	4	.625	.37	2.500	1.900	2.816
7″	55	3½	.507	.35	1.775	1.949	2.100
6″	50	3½	.531	.31	1.858	1.284	2.072
6″	40	2¾	.517	.25	1.421	1.158	1.614
5″	36	3	.400	.30	1.200	1.200	1.400
5″	30	2¾	.375	.25	1.000	1.000	1.166
4″	30	2¾	.410	.25	1.135	.730	1.257
4″	18	2	.281	.21	.562	.682	.676

Depth.	Weight per yard.	EFFECTIVE DEPTH.		Load Factor $8D\left(a+\frac{a'}{6}\right)S$ When $S=6$ Tons.	Deflection Factor $\left(a+\frac{a'}{6}\right)d^2$
		D feet	d feet		
15″	200	1.150	13.80	410	1415
15″	150	1.170	14.04	302	1062
12″	170	.910	10.92	292	797
12″	125	.930	11.16	208	576
10½″	135	.800	9.62	178	478
10½″	105	.812	9.74	155	378
9″	150	.658	7.90	197	388
9″	84	.691	8.30	108	225
9″	70	.698	8.38	92	193
8″	81	.610	7.37	94	175
8″	65	.618	7.42	74	137
7″	69	.530	6.37	72	114
7″	55	.537	6.44	54	87
6″	50	.456	5.47	45	62
6″	40	.458	5.50	35	49
5″	36	.383	4.60	25	30
5″	30	.385	4.62	21	25
4″	30	.298	3.58	18	16
4″	18	.304	3.65	10	9

NOTES ON PRECEDING TABLE OF ROLLED I BEAMS.

The following remarks upon the table of Phœnix beams applies equally to rolled I beams of any manufacture:

Upper Half of Table.

The first column contains the total depth out to out of flanges in inches.

The second column contains the weights per yard length of beam.

The third column contains the width of flange in inches.

The fourth column contains the thickness of flange.

The fifth column contains the thickness of stem or web.

The sixth column contains the area of one flange.

The seventh column contains the area of stem.

The eighth column contains the sum of the area of one flange and 1-6 the area of stem.

Lower Half of Table.

Columns one and two, as before, contain the depths of beam in inches, and weights per yard run.

Column three contains the effective depth or separation of centers of gravity in feet; and column four the same function in inches.

Column five contains the factor for beam uniformly loaded, for maximum safe load, when $W =$ weight, is given in tons of 2000 pounds.

For safe center load take—

$$4\,D\left(a + \frac{a'}{6}\right)S$$

or one-half the values given in table. To illustrate, suppose a beam 15″ depth, 20 feet clear span supported at both ends, what is the safe equally distributed load. The load factor for this beam is 410, and—

$$W = \frac{410}{20} = 20.5$$

tons, and safe center load

$$W = \frac{205}{20} = 10.25$$

tons. Assuming ultimate tensile strength of beam as 60,000 pounds per square inch of section, then the breaking weights would be 102.5 tons for uniformly distributed load, and 51.25 tons for center load.

Column six contains the deflection factor thus, for above beam the deflection factor is 1415, and center deflection for load of 10.25 tons, is

$$D = \frac{.006 \times 10.25 \times 20^3}{1415} = .348''$$

and for uniformly distributed load of 20.5 tons, is

$$D = \frac{.004 \times 20.5 \times 20^3}{1415} = .4637''$$

COLUMNS.

Comparative strength of long columns or pillars from D. K. Clark's Manual for Mechanical Engineers:

Cast iron	1000
Wrought iron	1745
Cast steel	2518

The following are the celebrated Gordon formulae for the strength of cast iron columns:

For solid or hollow round columns,

$$W = \frac{40\,a}{1 + \dfrac{r^2}{400}}$$

For solid or hollow rectangular columns,

$$W = \frac{40\,a}{1 + \dfrac{r^2}{500}}$$

Where W = breaking load in tons of 2000 pounds, a = sectional area of metal in inches, and r = ratio of length to diameter of column. (In a taper column or columns of different diameters the least diameter is always considered in estimating the strength.) Of above breaking loads, from one-fourth to one-tenth may be allowed for safe working load, the largest factor of safety being employed when columns are subject to shocks or vibrations; a factor of safety of 4 being ample for quiescent loads.

The following formulae, by Messrs. Stoney, Unwin, and Baker, for wrought iron and steel columns are Gordon's formulae adapted to these materials:

For solid rectangular wrought iron columns,

$$W = \frac{17.92\,a}{1 + \dfrac{r^2}{3000}}$$

For columns of angle, channel tee or cruciform rolled iron,

$$W = \frac{21.28\,a}{1 + \dfrac{r^2}{900}}$$

For solid round columns of low grade steel,

$$W = \frac{33.6\,a}{\dfrac{r^2}{1400}}$$

For solid round columns of high grade steel,

$$W = \frac{57.12\,a}{1 + \dfrac{r^2}{800}}$$

For solid rectangular columns of low grade steel,

$$W = \frac{33.6\,a}{1 + \frac{r^2}{2480}}$$

For solid rectangular columns high grade steel,

$$W = \frac{57.12\,a}{1 + \frac{r^2}{1600}}$$

The following is the Gordon formula for breaking loads of pillars of white and yellow pine, based upon experiments by Mr. C. Shaler Smith:

$$W = \frac{2.5\,a}{1 + \frac{r^2}{250}}$$

An I beam of rolled iron, with squared ends, of following dimensions, depth of beam 12 inches, width of flange of 5.5 inches, length 24 feet, and area of cross-section 11.223 sq. inches, would require as a breaking load,

$$r = \frac{288}{5.5} = 52.36$$

and

$$W = \frac{21.28 \times 11.323}{1 + \frac{52.36^2}{900}} = 95 \text{ tons or a load}$$

of $\frac{59}{11.223} = 5.26$ tons or 10,520 pounds per sq. inch of section.

What is the breaking load of a round cast iron hollow column 18 feet long, with an internal diameter at smallest end of 8 inches, and an external diameter of 10.5 inches.

$$a = .7854\,(10.5^2 - 8^2) = 36.325 \text{ sq. inch}$$

$$r = \frac{18 \times 12}{10.5} = 20.57$$

and

$$W = \frac{40 \times 36.325}{1 + \frac{20.57^2}{400}} = 706 \text{ tons or a load}$$

of

$$\frac{706}{36.325} = 19.436 \text{ tons or } 38{,}872 \text{ pounds}$$

per sq. inch of section.

PIPER'S PATENT RIVETLESS COLUMNS.

THICKNESSES AND CORRESPONDING AREAS, AND WEIGHTS PER FOOT.

Thickness.	4 in. column.				6 in. column.				8 in. column.				10 in. column.				Thickness.
	4 Segments, incl'dg Battens		Weight of one Segment.	Weight of one Batten.	4 Segments, incl'dg Battens.		Weight of one Segment.	Weight of one Batten.	4 Segments, incl'dg Battens.		Weight of one Segment.	Weight of one Batten.	4 Segments, incl'dg Battens.		Weight of one Segment.	Weight of one Batten.	
Inch.	Area. Sq. in.	Wt. lbs.	lbs.	lbs.	Area. Sq. in.	Wt. lbs.	lbs	lbs.	Area. Sq. in.	Wt. lbs.	lbs.	lbs.	Area. Sq. in.	Wt. lbs.	lbs.	lbs.	Inch.
3-16	5.21	17.4	2.5		....	....	....	....	....	....	....	....	....	....	....	....	3-16
¼	6.00	20.0	3.1		7.30	24.3	4.2	1.87	10.98	36.6	6.1		....	....	....	....	¼
5-16	6.80	22.7	3.8	1.87	8.43	28.1	5.2		12.50	41.7	7.3		16.00	53.3	9.3		5-16
⅜	7.60	25.3	4.5		9.55	31.8	6.1		14.03	46.8	8.6		17.90	59.7	10.9		⅜
7-16	8.39	28.0	5.1		10.68	35.6	7.0	2.3	15.55	51.8	9.9	3.1	19.80	66.0	12.5		7-16
½	....	....	....	....	11.81	39.4	8.0		17.08	56.9	11.1		21.70	72.3	14.1		½
9-16	....	....	....	....	....	....	....	....	18.60	62.0	12.4		23.60	78.7	15.7	4.0	9-16
⅝	....	....	....	....	....	....	....	....	20.13	67.1	13.7		25.50	85.0	17.3		⅝
11-16	....	....	....	....	....	....	....	....	....	....	....	....	27.40	91.3	18.8		11-16
¾	....	....	....	....	....	....	....	....	....	....	....	....	29.30	97.7	20.4		¾

SHEARING RESISTANCE.

MATERIAL—		POUNDS PER SQ. IN.	
Steel, " different specimens		72,000 / 93,600	82,800
Wrought iron	Rankine.	50,000	
" Swedish bar	D. K. Clark.	42,112	
" $\frac{1}{8}''$ to $1\frac{1}{8}''$ bars	C. Little.	45,956	
Cast Iron	Rankine.	27,709	
"	Stoney.	19,040	
Hematite steel	Kirkaldy.	56,470	
Fagersta "	"	64,557	
Rivet iron	E. Clark.	54,096	
Ash and Elm	Rankine.	1,400	
Oak	"	2,300	
Red pine	"	,500 / ,800	,650
Spruce	"	,600	
Larch	"	,970 / 1,700	1,850

The resistance to shearing of links and pins varies as the square of the depth of the link and the square of the diameter of pin.

SHAFTING.

The following formulæ are adopted from Mr. D. K. Clark, for round shafting only;

Let D = transverse deflection in inches.
W = weight in pounds.
L = distance center to center of bearings in feet.
d = diameter of shaft in inches.
D' = angular deflection in degrees.
W' = twisting force in pounds.
R = radius of force in feet.
L' = length of shaft between couplings in feet.

Torsional Strength of Shafting—

Cast iron, $W' = \frac{373\, d^3}{R}$ $R = \frac{373\, d^3}{W}$ $d = \sqrt[3]{\frac{W R}{373}}$

Wrought iron, $W' = \frac{933\, d^3}{R}$ $R = \frac{933\, d^3}{W}$ $d = \sqrt[3]{\frac{W R}{933}}$

Steel, $W' = \frac{1120\, d^3}{R}$ $R = \frac{1120\, d^3}{W}$ $d = \sqrt[3]{\frac{W R}{1120}}$

Torsional Deflection of Shafting—

Cast iron, $D' = \dfrac{W' R L'}{11,100\, d^4}$

Wrought iron, $D' = \dfrac{W' R L'}{16,600\, d^4}$

Steel, $D' = \dfrac{W' R L'}{34,300\, d^4}$

The angle of torsion varies directly as the length of bar, but the torsional moment of rupture is independent of the length.

Mr. Clark regards a deflection of 1° in 20 diameters of length, as a good working limit, and suggests—

for cast iron shafts—

$$d = \sqrt[3]{\frac{W' R}{18.5}} \quad \text{and } W' R = 18.5\, d^3$$

for wrought iron—

$$d = \sqrt[3]{\frac{W' R}{27.7}} \quad \text{and } W' R = 27.7\, d^3$$

for steel—

$$d = \sqrt[3]{\frac{W' R}{57.2}} \quad \text{and } W' R = 57.2\, d^3$$

Transverse Deflection of Shafting.

	Supported at ends.	*Fixed at ends.*
Cast iron,	$D = \dfrac{W L^3}{39,400\, d^4}$	$D = \dfrac{W L^3}{79,900\, d^4}$
Wrought iron,	$D = \dfrac{W L^3}{66,400\, d^4}$	$D = \dfrac{W L^3}{133,000\, d^4}$
Steel,	$D = \dfrac{W L^3}{78,800\, d^4}$	$D = \dfrac{W L^3}{158,000\, d^4}$

The deflection should not exceed .01 inch per foot of length, or 1 inch in 100 feet; whence for shafts of—

	Supported at ends.	*Fixed at ends.*
Cast iron,	$d = \sqrt[4]{\frac{W L^2}{394}}$	$d = \sqrt[4]{\frac{W L^2}{790}}$
Wrought iron,	$d = \sqrt[4]{\frac{W L^2}{664}}$	$d = \sqrt[4]{\frac{W L^2}{1330}}$
Steel,	$d = \sqrt[4]{\frac{W L^2}{788}}$	$d = \sqrt[4]{\frac{W L^2}{1576}}$

Horse Power of Shafting.

Let S = revolutions per minute.
" H = horse power developed.

Cast iron round shafting—

$$18.5 \times 3.1416 \times 2 = 116.24 \text{ and } \frac{33000}{116.24} = 284$$

Wrought iron round shafting—

$$27.7 \times 3.1416 \times 2 = 174.04 \text{ and } \frac{33000}{174.04} = 189.6$$

Steel round shafting—

$$57.2 \times 3.1416 \times 2 = 359.4 \text{ and } \frac{33000}{359.4} = 91.82$$

Then—

$$\text{for cast iron,} \quad H = \frac{S d^3}{284}$$

$$\text{for wrought iron,} \quad H = \frac{S d^3}{189.6}$$

$$\text{for steel,} \quad H = \frac{S d^3}{91.82}$$

What power within safe limits will a round wrought iron shaft 2.5 inches diameter, transmit at 250 revolutions per minute.

$$H = \frac{250 \times 2.5^3}{189.6} = 20.6 \text{ horse power.}$$

A 10-inch engine shaft of wrought iron turns 80 times per minute, what is the power which it may transmit within safe limits?

$$H = \frac{80 \times 10^3}{189.6} = 421.94 \text{ horse power.}$$

STRENGTH OF STEEL SPRINGS.

Professor Rankine gives the following formula for the dimensions of helical steel springs:

Let D = diameter, or side of the square steel bar of which the spring is coiled—in 16ths of an inch.
W = load in pounds applied to the spring.
d = mean diameter of spring in inches.

Then—

$$D = \sqrt[3]{\frac{W d}{3}} \quad \text{for round steel.}$$

$$D = \sqrt[3]{\frac{W d}{4.29}} \quad \text{for square steel}$$

Mr. D. K. Clark quotes the following formulae for compression or extension of steel helical springs:

$$E \text{ or } C = \frac{d^3 W}{D^4 \, 22} \quad \text{for round steel.}$$

$$E \text{ or } C = \frac{d^3 W}{D^4 \, 30} \quad \text{for square steel.}$$

Where E = extension of one coil in inches,
C = compression of one coil in inches.
The extension or compression of one coil is to be multiplied by the number of coils for total deflection.

Mr. Clark also furnishes the following formulae for laminated steel springs:

$$E = \frac{1.482\, l^3}{b\, n\, t^3} \quad (1) \qquad \text{and } S = \frac{b\, n\, t^2}{10.09\, l} \quad (2)$$

$$l = \sqrt[3]{\frac{E\, b\, n\, t^3}{1.482}} \quad (3) \qquad \text{and } n = \frac{1.482\, l^3}{E\, b\, t^3} \quad (4)$$

$$l = \frac{b\, n\, t^2}{10.09\, S} \quad (5) \qquad \text{and } n = \frac{S\, l\, 10.09}{b\, t^2} \quad (6)$$

Where E = elasticity, or deflection, in 16ths of an inch per ton of 2000 pounds.
S = working strength, or load in tons of 2000 pounds.
l = span, when loaded, inches.

b — width of plates, in 16ths of an inch—supposed to be uniform.

t = thickness of plates, in 16ths of an inch.

n = number of plates.

NOTE *A*.—The span and elasticity are those due to spring *when loaded*.

NOTE *B*.—When extra thick back and short plates are used, they must be replaced, for the purpose of calculation, by an equivalent number of the ruling thickness, prior to application of equations (1) and (3). This is found by multiplying the number of extra thick plates by the cube of their thickness, and dividing by the cube of the ruling thickness. Conversely, the number of plates of the ruling thickness given by equation (4) required to be removed and replaced by a given number of extra thick plates, are found by the same calculation.

NOTE *C*.—It is assumed that the plates are similarly and regularly formed, and are of uniform width, and *slightly* tapered at the ends.

NOTE *D*.—Extra thick back or short plates must be replaced for the purpose of calculation, by an equivalent number of plates of the ruling thickness before applying equations (2) and (6). This is done by multiplying the number of extra thick plates by the square of their thickness, and dividing by the square of the ruling thickness. Conversely, the number of plates of ruling thickness given by equation (6) required to be removed and replaced by a given number of extra thick plates, are found by the same calculation.

STRENGTH OF STEAM BOILERS.

In marine and fire-box boilers, with flat surfaces, the resistance to rupture is measured by the strength of the stays and braces that hold the flat surfaces in shape. But in boilers with cylindrical shells the strength is measured by the thickness of the plate and the diameter; and the law of strength is expressed as follows:

$$P = \frac{t \times T}{D} \times 2,$$

where P = bursting pressure; t = thickness of plate;
T = tensile strength of the iron; D = diameter of shell.

Suppose a boiler 48-inch diameter of shell, of ¼-inch plate having a

tensile strength of 60,000 pounds per square inch of cross-section, the rupturing pressure would be,

$$P = \frac{.25 \times 60{,}000}{48} \times 2 = 625 \text{ pounds.}$$

Under United States inspection laws this boiler would be limited, for single riveted laps to one-sixth the maximum strength, or 104.16 pounds; but with double riveted laps, and holes drilled instead of punched, a working pressure of 125 pounds would be allowed.

The law of strength, as expressed in the formula, assumes a cylinder without a lap, but Fairbairn's experiments have shown that a ring or course, united by single riveted laps, possesses but .56 of the strength of a continuous ring, and with double riveted laps 25 per cent. additional strength, or .70 of strength of the continuous ring, These experiments, however, will not apply to a plate riveted into a boiler, as the width of course, is an element that materially affects the strength, and the strength of a shell is greater at the roundabout joint than in the solid plate. This fact is recognized in the United States inspection laws, and a working strain 1-6 to 1-5 the strength of the solid plate is allowed on single and double riveted and drilled laps, respectively.

The direction of greatest strain in a cylindrical steam boiler is at right angles to the axis. The strength of a steam boiler, in the direction of the axis, is represented by the formula,

$$P = \frac{D\,3.1416 \times T \times t}{D^2\ .7854}$$

Hence in a 48-inch boiler of ¼-inch plate, the strength in the direction of the axis is

$$P = \frac{48 \times 3.1416 \times .25 \times 60{,}000}{48^2 \times .7854} = 1250 \text{ pounds}$$

$$\text{and } \frac{1250}{625} = 2,$$

Thus the strength of a boiler in the direction of the axis, is twice the strength at right angles to the axis. Or, in other words, the strain on the roundabout seams is but one-half the strain on the longitudinal seams.

At the roundabout joint there is one force tending to pull the courses apart, and one force tending to tear the joint parallel with the axis, but the resistance to this latter force is two thicknesses of plate

instead of one. Assuming 56 per cent. as the strength of the single riveted joint, the roundabout joint possesses a strength of 1.12 as compared with the solid plate, for the circumferential resistance to rupture, but a strength of .56 as compared with the solid plate for the resistance to rupture in the direction of the axis.

If the separation of the roundabout seams was infinity, the strength of a course single riveted would be .56 of the solid plate, but if the separation was 0, the strength of a course would be 1.12 of the solid plate. As the distance between the roundabout seams diminishes, the co-efficient of strength increases. Hence it appears that narrow sheets are peferable to wide ones when a boiler is to be made up in courses; and that a boiler of courses with one sheet to a course, is no stronger than with two or more sheets to a course.

The strength of flues is expressed by the following formula, deduced from Mr. Fairbairn's experiments on the collapsing pressure of tubes:

$$P = K\frac{t^{2\cdot19}}{L\,D} \text{ whence } \frac{P}{K} = \frac{t^{2\cdot19}}{L\,D} \text{ therefore,}$$

$$t^{2\cdot19} = \frac{P\,L\,D}{K} = \text{and } t = \sqrt[2.19]{\frac{P\,L\,D}{K}}$$

Where P = collapsing pressure;
K = a constant deduced by Fairbairn as 806,300;
t = thickness of flue or tube;
L = length of flue in feet;
D = diameter of flue in inches; (2 is usually substituted for 2.19 as the power of the thickness.)

From this it appears that the resistance to collapse of flues varies directly as the 2.19 power of the thickness, inversely as the length, and inversely as the diameter.

Experience has shown that the roundabout laps of flues contribute to the resisting power, but precisely in what ratio has not been determined. Fairbairn suggests that a flue 6 feet long, made in three lapped courses, is equivalent in strength to a flue one-third the length, or 2 feet, and that a flue made of three or more courses should be involved in the equation at ⅓ its length.

Example: Boiler 24 feet long, flues 20 inches diameter, working pressure 104.16 pounds, factor of safety 4, desired thickness of flue, if made of courses,

$$\frac{24}{3} = 8, \text{ reduced length.}$$

Collapsing pressure, $104.16 \times 4 = 416.64$ pds.

$$\text{hence } t = \sqrt{\frac{116.64 \times 8 \times 20}{806{,}300}} = .2875''$$

WEIGHT OF ROUND, SQUARE AND PLATE IRON PER FOOT.

Diameter or Thickness.	W'ght 1 foot sq.	W'ght round	W'ght sq'ure	Diameter or Thickness.	W'ght 1 foot sq.	W'ght round	W'ght sq'are
1-32 = .0312	1.263	.0026	.0033	3 5/8 = 3.625	146.5	34.836	44.418
1-16 = .0625	2.526	.010	.013	3 3/4 = 3.75	151.6	37.332	47.534
1/8 = .125	5.052	.041	.053	3 7/8 = 3.875	156.6	39.864	50.756
3-16 = .1875	7.578	.093	.119	4	161.7	42.464	54.084
1/4 = .25	10.104	.165	.212	4 1/8 = 4.125	166.7	45.174	57.517
3/8 = .375	15.160	.373	.475	4 1/4 = 4.25	171.8	47.952	61.055
1/2 = .5	20.208	.663	.845	4 3/8 = 4.375	176.8	50.815	64.700
5/8 = .625	25.260	1.043	1.320	4 1/2 = 4.5	181.9	53.760	68.448
3/4 = .75	30.312	1.493	1.901	4 5/8 = 4.625	186.9	56.788	72.305
7/8 = .875	35.370	2.032	2.588	4 3/4 = 4.75	192.0	59.900	76.264
1	40.420	2.654	3.380	4 7/8 = 4.875	197.0	63.094	80.333
1 1/8 = 1.125	45.470	3.360	4.278	5	202.1	66.752	84.480
1 1/4 = 1.25	50.520	4.172	5.280	5 1/8 = 5.125	207.1	69.731	88.784
1 3/8 = 1.375	55.570	5.020	6.390	5 1/4 = 5.25	212.2	73.172	93.168
1 1/2 = 1.5	60.630	5.972	7.604	5 3/8 = 5.375	217.2	76.700	97.657
1 5/8 = 1.625	65.680	7.010	8.926	5 1/2 = 5.5	222.3	80.304	102.24
1 3/4 = 1.75	70.730	8.128	10.325	5 5/8 = 5.625	227.3	84.001	106.95
1 7/8 = 1.875	75.780	9.333	11.883	5 3/4 = 5.75	232.4	87.776	111.75
2	80.840	10.616	13.520	5 7/8 = 5.875	237.5	91.634	116.67
2 1/8 = 2.125	85.890	11.988	15.263	6	242.5	95.552	121.66
2 1/4 = 2.25	90.940	13.440	17.112	6 1/4 = 6.25	252.6	103.70	132.04
2 3/8 = 2.375	95.990	14.975	19.066	6 1/2 = 6.5	262.7	112.16	142.82
2 1/2 = 2.5	101.00	16.688	21.120	6 3/4 = 6.75	272.8	120.96	154.00
2 5/8 = 2.625	106.10	18.293	23.292	7	282.9	130.05	165.63
2 3/4 = 2.75	111.20	20.076	25.560	7 1/2 = 7.5	303.0	149.33	190.14
2 7/8 = 2.875	116.20	21.944	27.939	8	323.3	169.85	216.34
3	121.30	23.888	30.416	8 1/2 = 8.5	343.5	191.81	244.22
3 1/8 = 3.125	126.30	25.926	33.010	9	363.8	215.04	273.79
3 1/4 = 3.25	131.40	28.040	35.704	10	404.2	266.30	337.92
3 3/8 = 3.375	136.40	30.240	38.503	12	485.0	382.21	485.00
3 1/2 = 3.5	141.50	32.512	41.408				

For steel	multiply	by	1.01
" copper	"	"	1.125
" lead	"	"	1.47
" brass	"	"	1.06
" zinc	"	"	0.9
" tin	"	"	0.95
" cast iron	"	"	0.928

The weight of iron (and other materials) depends upon the purity—homogeneity—of the ore from which it is made—and whether hammered or rolled. The table is for rolled iron. And the weights of plate iron are based on uniform thickness. The spring of the rolls in the center makes the average weight somewhat greater.

The weight of bar iron up to 12″ wide and 12″ thick, can be readily obtained from the above table. Suppose we want the weight of $2\frac{1}{2} \times \frac{1}{2}$ in flat bar. The weight of $2\frac{1}{2} \times 2\frac{1}{2}$ inch bar is 21.120, and

$$2\frac{1}{2} \times \frac{1}{2} = \frac{21.120}{5} = 4.224 \text{ pds.}$$

Suppose we want the weight of $5 \times \frac{1}{4}$. The weight of $5 \times 5 = 84.480$ and $\frac{5}{.25} = 20$, hence $\frac{84.480}{20} = 4.224$ pds.

THICKNESS OF CAST IRON WATER PIPE.

The following formula adapted from Neville, is believed to be a safe equation for the thickness of cast iron pipe for public water supply:

$$t = \frac{9}{S}\left[.0016\left(\frac{h}{33} + 10\right) d\right] + .32$$

Where t = thickness of pipe in inches,
h = head or pressure in feet,
d = diameter of pipe in inches,
S = the tensile strength of metal in tons of 2000 pounds.

What should be thickness of a 20-inch water main subject to a maximum pressure of 150 pounds per square inch, or $150 \times 2.308 = 346.2$ feet head, with cast iron of 18000 pounds tensile strength.

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 20\right] + .32 = .9757''.$$

What should be the thickness of 40-inch pipe for same service and of same metal,

$$t = \frac{9}{9} \times \left[.0016\left(\frac{346.2}{33} + 10\right) \times 40\right] + .32 = 1.6313''.$$

WEIGHTS OF CAST IRON WATER PIPES.

In pounds per foot run including bells and spigots.

Diameter.	Philadelphia.	Chicago.	Cincinnati.		Standard	Light.
			Weight.	Th'ckn's		
2 inch	—	—	—	—	7	6
3 "	15.000	—	17	½"	15	13
4 "	21.111	21.167	23	"	22	20
6 "	30.106	36.666	50	¾"	33	30
8 "	40.683	50.000	65	"	42	40
10 "	52.075	65.000	80	"	60	55
12 "	69.162	83.333	100	"	75	70
16 "	102.522	125.000	130	"	—	—
20 "	147.681	—	200	⅞"	—	—
24 "	—	250.000	224	"	—	—
30 "	—	—	300	1"	—	—
36 "	—	450.000	430	1⅛"	—	—

Water-pipe is usually tested to 300 pounds pressure per square inch before delivery; and a hammer test should be made while the pipe is under pressure.

The Philadelphia lengths for each section are for 3 and 4 inch pipe, 9 feet. All larger sizes 12 feet 3¼ inches in length.

The Cincinnati lengths are uniform for all diams. 12 feet.

Chicago same as Cincinnati.

Standard lengths are for 2 inch pipe, 8 feet; and all other sizes 12 feet.

THICK CYLINDERS.

For cylinders where the thickness is small compared with the diameter the formula for strength of steam boiler shells will apply. Let P = rupturing pressure, t = thickness of plate, D = diameter of cylinder, and T = the tensile strength of the material.

Then—

$$P = \frac{t\,T \times 2}{D} \text{ whence}$$

$$D = \frac{t\,T \times 2}{P} \text{ and } t = \frac{D\,P}{T \times 2}$$

But when the thickness of cylinder (as in hydraulic presses), becomes large as compared with diameter, then the following formula applies:

$$\frac{R}{r} = \sqrt{\frac{T + P}{T - P}} \text{ and}$$

$$P = T\,\frac{R^2 - r^2}{R^2 + r^2} \text{ whence } R = r\sqrt{\frac{T + P}{T - P}}$$

When R = radius outer circumference, r = radius inner circumference, T = tensile strength of the material, and P = maximum pressure, which is usually five to eight times the working pressure.

Suppose a cylinder 8″ internal diameter, 4″ thick, of cast iron, having a tensile strength of 16,500 pounds; desired bursting pressure. Inner radius 4″, outer radius 8″. Hence,

$$P = \frac{8^2 - 4^2}{8^2 + 4^2}\,16{,}500 = \frac{48}{80}\,16{,}500 = 9{,}900 \text{ pounds.}$$

M. Lamè has pointed out the important fact that when the internal pressure in a cylinder is equal to or greater than the co-efficient of strength of the material, no thickness, however great, will enable the cylinder to withstand the pressure. Thus, let P = the tensile resistance of cast iron = 16,500 pounds. Then, by equation,

$$\frac{R}{r} = \sqrt{\frac{16{,}500 + 16{,}500}{16{,}500 - 16{,}500}} = \frac{33{,}000}{0} = \infty$$

It will be observed from this demonstration that no matter what may be the value of "r," R will be infinitely greater.

In designing hydraulic presses it is customary to give the ram such a diameter as will develop the required maximum pressure without overstrain of the cylinder. Thus, suppose a press with an 8″ ram to exert 150 tons maximum pressure, the area of an 8″ ram is 50 sq. in. Hence, pressure per sq. in. of ram to exert 150 tons:

$$\frac{150}{50} 2{,}000 = 6{,}000 \text{ pds.},$$

and the thickness of such a cylinder of cast iron with a factor of safety of 2 would be

$$R = 4\sqrt{\frac{16{,}500 + 12{,}000}{16{,}500 - 12{,}000}} - 4 = 6.064''$$

A manufacturer of hydraulic machinery in this city (Cincinnati) contracted to furnish the American Pressed Tan Bark Company, N. Y., a compress for baling pulverized bark, which should with safety produce a maximum pressure of 1,500 tons on the ram and bale. As 1,500 tons was a constant working load, the factor of safety should have been not less than 4, and in view of the expensive character of the machinery a factor of safety of 6 was preferable.

The ram was 20.05 inches diameter = 315.733 sq. inches area, and pressure per sq. inch equivalent to 1,500 tons load is

$$\frac{1{,}500 \times 2{,}000}{315.733} = 9{,}501.7 \text{ pounds.}$$

The external diameter of cylinder was 45 inches and internal diameter 21.9375 inches, whence

$R = 22.5$ inches, and $r = 10.9687$ inches.

$T =$ may be taken at 20,000 pounds for first class car wheel iron, then

$$P = 20{,}000 \frac{22.5^2 - 10.9687^2}{22.5^2 + 10.9687^2} = 12{,}319.2 \text{ pounds,}$$

and a factor of safety of

$$F_s = \frac{12{,}319.2}{9{,}501.7} = 1.296 \quad \text{instead of 4 or 6.}$$

The safety valve which was furnished for the press and said to represent a maximum load on ram of 1,500 tons, contained the following elements. (See Safety valves.)

$L = 22.8125$ inches.	$L' = 1.15625$ inches.
$L'' = 9.86$ inches.	$W = 74$ pounds.
$w = 2.77$ pounds.	$w' = 2$ pounds.
$a = .3167$ sq. inches.	

and pressure per sq. inch represented by safety valve with weight in extreme notch of lever,

$$p = \frac{\frac{74 \times 22.8125}{1.15625} + \left(\frac{2.77 \times 9.86}{1.15625} + 2\right)}{.3167} = 4{,}691 \text{ pounds per sq. inch}$$

of ram, or

$$\frac{4{,}691 \times 315.733}{2{,}000} = 740.552 \text{ tons load on ram, or less than}$$

one-half the contract pressure on the bale.

THICK HOLLOW SPHERES.

Let R = external radius.
r = internal radius.
S = tensile strength in pounds per sq. inch of section of the material, and
P = bursting pressure.

Then—

$$P = \frac{S(2R^3 - 2r^3)}{R^3 + 2r^3}$$

$$R = r\sqrt[3]{\frac{2(S+P)}{2S-P}} \text{ and } r = \frac{R}{\sqrt[3]{\frac{2(S+P)}{2S-P}}}$$

In thick spheres (as in thick cylinders), it appears that when the pressure $P = 2S$, that no thickness however great will resist the strain.

Let r = internal radius = 5 inches.
S = tensile strength of cast iron = 18,000 pounds.
P = 36,000 pounds per sq. inch, then

$$R = 5\sqrt[3]{\frac{2(18000 + 36000)}{2 \times 18000 - 36000}} = 5\sqrt[3]{\frac{108000}{0}} = \infty$$

Let r = 5 inches.
R = 9 inches.
S = 18,000 pounds,

desired the bursting pressure of such a shell.

$$P = 18{,}000 \frac{(2 \times 9^3) - (2 \times 5^3)}{9^3 + (2 \times 5^3)} = 22{,}210.4 \text{ pounds per sq. inch.}$$

and

$$R = 5 \sqrt[3]{\frac{2(18{,}000 + 22{,}210.4)}{2 \times 1{,}800 - 22{,}210.4}} = 9 \text{ inches. and}$$

$$r = \frac{9}{\sqrt[3]{\frac{2(18{,}000 + 22{,}210.4)}{2 \times 18{,}000 - 22{,}210.4}}} = 5 \text{ inches.}$$

STEAM BOILER EXPLOSIONS.

No general cause can be cited for steam boiler explosions; but a careful analysis of all the facts will generally enable the experienced engineer to arrive at a probable cause, in nearly every instance.

Low water is rarely the cause of an explosion, except in fire-box boilers, where the crown of the furnace (which is subjected to the highest temperature) is uncovered and crushed in. But in boilers fired under the shell, with return tubes or flues, it is extremely doubtful if low water is ever the cause of an explosion.

Low water, when it is sufficiently low to permit overheating of the plates below the fire line, may, and in many cases does, contribute to weaken the boiler. When the expansion is in excess of the thermo-elastic limit of the iron, a permanent set occurs, and the iron is in precisely the same condition as though the limit of elasticity had been exceeded by overstrain.

Initial strain is more frequently the cause of explosion than is generally supposed. Many boilers made of good iron, are put together in such a haphazard and reckless manner that the factor of safety with which they are worked, instead of being 5 or 6, may be but a trifle in excess of the working pressure. A boiler of this kind, after suffering the deterioration due to a limited use, is very liable to rupture and explosion at, or even below the working pressure, and occasionally they let go in the shop under trial.

Overpressure—this was Mr. Fairbairn's theory of explosion; but instances have been noted where violent explosions have occurred at less than the working pressure; and with the usual pressure and

safety-valve blowing, boilers have let go. Overpressure, however, in connection with excessive initial strain, is a fruitful source of disaster in the use of steam boilers. Defective steam-gauges, although a trifling detail in themselves, have contributed to ruptures and explosions by false indications. Safety-valves are generally set to blow by the steam-gauge, and when this is an unreliable device (which is the rule rather than the exception), then the safety-valve becomes a delusion.

Explosions sometimes happen when boilers filled with comparatively cold water and cold themselves, are incautiously fired.

When the *regime* of a steam boiler is fully established, all parts of the shell and flues or tubes are practically at the same temperature, and forcing the fires is less liable to work injury; but when a boiler is filled with cold water, and fires are started after an interum of idleness, the rapid firing has the effect of subjecting the bottom of the boiler to an expansion corresponding to the elevation of temperature, while the top of the boiler is yet cold. The strains, by reason of the extra expansion of the bottom of the boiler, may be, and in some cases are, sufficient to produce incipient fractures of plates or joints, and place the boiler in condition for a violent explosion, at less than the working pressure.

Overheating of the iron and water is no doubt responsible for certain explosions. So long, however, as the water is in contact with the plate, it is difficult to produce an overheat of the iron; but when the water is repelled or "lifted" from the plate an instant of time is sufficient to produce a dangerous overheat in the courses nearest the fire. This overheat not only subjects the boiler to the strains of excessive expansion, but materially reduces the cohesive strength of the iron, in addition to which a proportionally large evaporation takes place when the water returns to the plates.

It is well known that when water is deprived of air, it can be elevated to a temperature higher than the boiling point before vaporization occurs. M.M. Donney and Magnus have made experiments on ebullition under the pressnre of the atmosphere, and the former found that by carefully freeing the water of air, he could elevate the temperature to 275 degrees Fahr., before vaporization occurred, and when it did occur, the action was not like ordinary ebullition under pressure of the atmosphere, but was instantaneous and explosive, a portion of the water being violently projected from the test tubes.

The temperature (275 F.) corresponds to a pressure of about three atmospheres, and M. Donney concludes that this pressure is equivalent to the natural force of cohesion of the particles of water.

How far the results obtained by Donney and Magnus may be used to solve the problem of steam boiler explosions, is not known. But there can be no doubt that similar and instantaneous evaporation often takes place in a steam boiler, and whether the effect is to produce a rupture. simply depends upon the strength of boiler and quantity of water acted upon.

The theory of repulsion, so ably argued by Mr. Robinson, is perhaps the most plausible for those explosions with the usual level of water in the boiler and every indication that no danger exists. Experience has shown that when the iron of a boiler otherwise clean, is heated to a temperature of 380 to 420 deg. Fahr., the water is repelled from the plate, and under this condition the iron of the boiler may be heated to the temperature of the impinging hot gas, Whenever the equilibrium within the boiler is destroyed, the water returns to the hot plates, and a large and instantaneous evaporation occurs. This, instead of naturally passing through the superincumbent water, carries the water with it, and projects it against the bounding surfaces of the boiler. If the mechanical effect of this percussive action be sufficient to produce a rupture, then there is an immediate reduction of pressure, followed by a further and larger evaporation, which, in seeking to escape, rushes through the vent with a velocity proportional to the unbalanced pressure, and carries the now dismembered boiler with it, upon the same principle that a mountain torrent can convey large rocks for great distances, and a whirlwind carry for miles bodies of matter having a greater specific gravity than the air.

Engineers are generally united in the opinion that the most disastrous explosions are those occurring with boilers carrying the usual level of water, and that the violence of the explosion is directly proportional to the weight of water in the boiler at time of rupture.

Corrosion, internal scale and deposits, improper setting, impeded circulation, and improper steam and water connections between batteries of boilers, have each contributed to swell the list of explosions.

With our existing knowledge of steel and iron plate, and with honest construction, there is no need of disastrous explosions in the use of steam boilers at the present time. If all the requirements are first known, any intelligent mechanical engineer can design a boiler or system of boilers which will not only comply with all other proper conditions, but will be absolutely safe as against violent explosion.

SPECIFIC GRAVITY.

	Specific Gravity.	Weight per cu. ft.
Water at 62° Fahr	1.000	62.321
Metals.		
Platinum	21.522	1342.000
Gold	19.425	1205.000
Mercury	13.596	848.750
Lead	11.418	712.000
Silver	10.505	655.000
Bismuth	9.900	616.978
Copper, hammered	8.917	556.000
" sheet	8.805	549.000
" cast	8.600	537.000
Gun metal, 84 copper, 16 tin	8.560	533.468
" 83 " 17 "	8.460	527.235
Nickel, hammered	8.670	540.223
" cast	8.280	516.018
Bearing metal, 79 copper, 21 tin	8.730	544.062
Brass, wire	8.540	533.000
" cast, 75 copper, 25 zinc	8.450	526.612
" " 66 " 34 "	8.300	517.264
" " 60 " 40 "	8.200	511.032
Bronze	8.400	524.000
Steel	7.852	490.000
Iron, wrought, average	7.698	480.000
" cast, "	7.110	444.000
Zinc, sheet	7.200	449.000
" cast	6.860	424.000
Tin	7.409	462.000
Antimony	6.710	418.174
Iron ores	5.251 3.829	327.247 238.627
Aluminum, cast	2.560	159.542
Minerals, Masonry, etc.		
Manganese	8.00	498.568
Basalt	3.00	187.000
Glass, flint	3.00	187.000
" plate	2.70	169.000
Marble	2.84 2.52	176.991 157.049
Granite	3.06 2.36	190.702 147.077
Soapstone, steatite	2.73	140.000
Flint	2.63	164.200
Feldspar	2.60	162.300
Limestone	2.8 2.7	175.000 169.000
Slate	2.90 2.80	181.000 175.000

Minerals, Masonry, etc.	Specific Gravity.	Weight per cu. ft.
Trap rock	2.72	170.000
Quartz	1.26 2.65	78.524 165.000
Shale	2.60	162.000
Sandstone, average	2.30	144.000
Gypsum, Plaster of Paris	2.30	144.000
Masonry	2.30 1.85	144.000 116.000
Graphite	2.20	137.106
Brick	2.167 2.000	135.000 125.000
Chalk	2.78 1.87	174.000 117.000
Sulphur	2.00	125.000
Clay	1.92	120.000
Sand, damp Gravel	1.9	118.000
" dry Gravel	1.42	88.600
Marl	1.90 1.60	119.000 100.000
Mud	1.63	102.000
Coal, anthracite	1.602	100.000
" bituminous	1.44 1.24	89.900 77.400
Coke, dry, loose, average	0.449	28.000
Scoria	0.83	51.726
Cement, American, Rosendale, loose	....	60.000
" " " well shaken	...	70.000
" " " thor'ly shaken.	....	80.000
" " struck bushel, 75 pounds	....	... ..
Liquids.		
Acid, sulphuric	1.840	114.670
" nitric	1.220	76.031
" acetic	1.080	67.306
Milk	1.030	64.100
Sea water	1.026	64.050
Linseed oil	0.940	58.680
Sperm oil	0.923	57.620
Olive oil	0.915	57.120
Alcohol, proof spirit	0.920	57.335
" pure	0.791	49.380
Petroleum	0.878	54.810
Turpentine, oil	0.870	54.310
Naphtha	0.848	52.940
Ether	0.716	44.700
Timber.		
Ash	0.753	47.0
Bamboo	0.400	25.0
Beech	0.690	43.0
Birch	0.711	44.4
Blue Gum	0.834	52.5
Boxwood	0.960	60.0
Cedar of Lebanon	0.486	30.4
Cherry, dry	0.672	42.0
Chestnut	0.535	33.4

Timber	Specific Gravity.	Weight per cu. ft.
Cork	0.250	15.6
Ebony, West India	1.193	74.5
Elm	0.544	34.0
Greenheart	1.001	62.5
Hawthorn	0.910	57.0
Hazel	0.860	54.0
Hemlock, dry	0.400	25.0
Holly	0.760	47.0
Hickory	0.850	53.0
Hornbeam	0.760	47.0
Laburnum	0.920	57.0
Lancewood	1.010 0.675	63.0 42.0
Lignum Vitæ	1.330 0.650	83.0 41.0
Locust	0.710	44.0
Mahogany, Honduras	0.560	35.0
" Spanish	0.850	53.0
Maple	0.790	49.0
Oak, live, dry	0.950	59.3
" white, dry	0.830	51.8
Pine, white, dry	0.400	25.0
" yellow, dry	0.550	34.3
" Southern, dry	0.720	45.0
Sycamore	0.590	37.0
Teak, Indian	0.880 0.660	55.0 41.0
Water Gum	1.001	62.5
Walnut	0.610	38.0
Willow	0.400	25.0
Yew	0.800	50.0
Miscellaneous.		
Ivory	1.82	114.000
India rubber	0.93	58.000
Lard	0.95	59.800
Gutta Percha	0.98	61.100
Beeswax	0.97	60.500
Turf, dry, loose	0.401	25.000
Pitch	1.15	71.700
Fat	0.93	58.000
Tallow	0.936	58.396

Gases.

Weight per cubic foot at 32° Fahr. and under pressure of one atmosphere:

Air	0.080728
Carbonic acid	0.12344
Hydrogen	0.005592
Oxygen	0.089256
Nitrogen	0.078596
Steam (ideal) Rankine	0.05022
Vapor of Ether, Rankine (ideal)	0.2093
" " Bi-sulphide of carbon, Rankine	0.2137
Olefiant gas (marsh gas)	0.0795

EXPLOSIONS IN FLOUR MILLS.

The recent explosion in the Washburn Mills at Minneapolis, together with the explosion of a similar nature (some six years ago) in the Tradeston Mills, Glasgow, Scotland, have awakened an inquiry among millers, as to the probable cause and means to prevent a recurrence of these wholesale disasters.

Prof. Rankine (whose judgment upon a question of this nature is practically above criticism) investigated the Glasgow accident, and, after mature consideration, advanced the opinion that the explosion (so-called) was due to the rapid ignition of combustible matter in the exhaust box, the fire traveling through the box into the dust room, the contents of which, were combustible matter in a finely comminuted state, moisture, and atmosphere. The dust room of the Tradeston Mills was located in the mill building; and the expansive effect of the inflamed carbon, evaporated moisture, and highly-heated air, in any but a very open room would be sufficient to raze the walls and communicate fire to the remainder of the building. The feed going off a pair of stones, the flinty buhrs struck fire and furnished the means of ignition of the matter in the exhaust box.

Experiments have been made on the combustion of finely-divided charcoal, and on dust from wood-working establishments; and when these substances are showered over a flame, the combustion is as instantaneous as alcohol or a hydro carbon.

When a finely comminuted carbonaceous substance is ignited, the instantaneous expansion of the ambient atmosphere is similar to that of the burning of a loose charge of powder, and when this combustion occurs in a tight dust room it is not difficult to anticipate the effect.

Mr. W. L. Barnum, Secretary of the Millers' National Insurance Company, furnishes the author the following facts in relation to the explosion at the Washburn Mills: "The dust in large mills is stored and sold, but in small establishments, the daily quantity is too insignificant to justify storage, and it is usually blown out of the mill. At the Washburn Mill the daily yield was about 3000 pounds, and worth $16.00 per ton of 2000 pounds or $24.00 per day. This dust, having a a lower specific gravity than the meal, was drawn by a carefully-adjusted pneumatic exhaust from the usual spouts into a tight dust room in the basement of the mill. In the transit from the buhrs to the dust room this material passed through an exhaust fan; hence

from the fan to the buhrs a partial vacuum subsisted, while from the fan to the dust room the air was appreciably compressed." Compressed air having a greater density than the normal atmosphere, the dust was readily held in mechanical suspension, and the air in this room was continually charged with a large percentage by volume of this finely divided matter. Under these conditions it is only necessary that the dust be combustible to produce what is termed the explosion.

Experiments have been made, according to Mr. Barnum, to prove that when this matter is showered into a close atmosphere it is consumed with a flash like gunpowder, and the natural expansion of the investing atmosphere, in the close dust room, due to the instantaneous elevation of temperature, would be sufficient to rend the strongest walls and communicate the flame to the mill building.

This fine dust, being almost entirely carbon, would ignite with the rapidity of a gas, which it practically was, in its thorough dissemination through the atmosphere; and if this material contained by absorption a quantity of moisture, the expansive effect would be greatly increased, as each cubic inch of water would occupy a cubic foot when converted into steam under the pressure of an atmosphere.

It is, therefore, not necessary to assume the generation of a specific gas having the property of instantaneous ignition, to account for these explosions; nor to assume the presence of olefiant gas (as some one has suggested), which is of spontaneous generation in certain localities, as all the elements necessary to a first-class disaster are present under the conditions of pneumatic exhaust and tight dust room.

COMBUSTION.

A certain energy is always expended in effecting the chemical combination of two or more elements, and this energy is exactly accounted for by the resultant heat.

The heat developed by the combination of oxygen—with carbon and hydrogen, is that employed in the mechanic arts. The chief constituents of fuel are carbon and hydrogen, and the union of oxygen with these elements, we term combustion. When the combustion is rapid, it is termed burning, when it is slow it is termed decomposition.

The temperature of combustion depends upon the rapidity with

which the combination is effected, but the heat developed by combustion is independent of the time, and depends only upon the calorific value of the element with which the oxygen combines.

The atmosphere, from which source the oxygen is obtained to support combustion, is composed of oxygen and nitrogen in mechanical combination, in the proportion of 8 atoms of oxygen to 28 atoms of nitrogen. Or, as more elegantly expressed in chemical terms, one equivalent of oxygen to two of nitrogen. The nitrogen is inert, and neither assists nor retards combustion.

When one pound of carbon unites with one and one-third pounds of oxygen, carbonic oxide is formed, and combustion is said to be imperfect or incomplete. Thus, to produce carbonic oxide, there are required one equivalent of carbon (6), and one equivalent of oxygen (8), and CO is the result.

When one pound of carbon unites with two and two-thirds pounds of oxygen, carbonic acid is formed, and combustion is said to be perfect, or complete. Thus, carbonic acid is composed of one equivalent of carbon (6), and two equivalents of oxygen (16), and CO_2 is the result.

When one pound of hydrogen combines with eight pounds of oxygen, vapor of water is formed. Thus water, or steam, consists of one equivalent of hydrogen and one equivalent of oxygen, and HO is the result.

According to the deductions of M. M. Favre and Silberman, the total heat of combustion of one pound of hydrogen when burned to vapor of water is 62,032 British thermal units, and the total heat of combustion of one pound of carbon, when burned to carbonic oxide, is 4,400 thermal units. The total heat of combustion of one pound of carbon burned to carbonic acid is 14,500 thermal units.

The air required for combustion can be determined as follows: It has been shown that when two equivalents of oxygen unite with one equivalent of carbon, carbonic acid is the result. Now, air consists of oxygen and nitrogen in the proportions of $\underline{8\ O}$ to $\underline{28\ N}$, and carbonic acid consists of one atom of carbon to two $\overline{\text{and}}$ $\overline{\text{two-thirds}}$ atoms of oxygen. Hence, to burn one pound of carbon to carbonic acid there is required of air

$$\frac{8 + 28 \times 2\frac{2}{3}}{8} = 12 \text{ pounds.}$$

Prof. Johnson's exhaustive experiments on coals for the U. S. Navy have shown that with natural draft of furnace, the theoretical quantity of air is insufficient for complete combustion, and that twice this amount is really required.

The specific gravity of air as compared with water is $\frac{1}{815}$ at temp.

of 60° Fahr.. and pressure of one atmosphere (1.47 pounds), and a cubic foot of water at same temp. and pressure, according to Berzelius, is 62.331 pounds. Hence, minimum volume of air required for one pound of carbon burned to carbonic acid becomes

$$\frac{12 \times 815}{62.331} = 157 \text{ cubic feet.}$$

The temperature of combustion has not been determined by direct experiment, but, as suggested by Prof. Rankine, may be calculated by dividing the calorific power or total heat of combustion of one pound of the combustible, by the weight into the specific heat of the products of combustion. We have seen that twelve pounds of air are necessary to produce two and two-thirds pounds of oxygen. Hence, the weight of products of combustion of one pound of carbon is thirteen pounds (carbonic acid 3⅔ pounds, nitrogen 9⅓ pounds.) The specific heat of carbonic acid, according to Regnault, is .2164, and of nitrogen .244. Hence, mean specific heat of products of combustion:

$$\frac{(3.66 \times .2164) + (9.33 \times .244)}{13} = .236$$

and $\frac{14{,}500}{13 \times .236} = 4\,574°$ Fahr. the resultant elevation of temperature.

But experience has shown that as much air is required for dilution as for combustion; hence, $12 \times 2 = 24$ pounds of air; and weight of products of combustion become for one pound of carbon burned to carbonic acid—air 12, nitrogen 9⅓, carbonic acid 3⅔ = 25 pounds and mean specific heat $\frac{(.236 \times 13) + (.238 \times 12)}{25} = .237$, and elevation of temperature becomes

$$\frac{14{,}500}{25 \times .237} = 2{,}447.7° \text{ Fahr.}$$

.238 is the specific heat of dry air, according to Regnault.

The temperature may be taken experimentally by calorimetric process as described in the section of this Manual devoted to Heat, for which purpose rods of iron, steel, or platinum are subjected to the temperature of the impinging hot gas in the fire chamber, over the bridge wall, in the back connection, or in the uptake for such a length of time as will permit them to acquire the full temperature, and are then quickly cooled down in a known weight of water.

For temperatures below 800 Fahr. a metal pyrometer will furnish fair approximations.

COMPOSITION OF FUEL

Charcoal, coke, coal, wood and peat, are the fuels principally in use. Charcoal is obtained by eliminating the volatile matter from wood or peat by distillation in a retort, or by partial combustion in a heap. A larger yield of carbon is obtained by the distillation process. According to Peclet, charcoal consists of carbon 93 per cent., and non-combustible or ash 7 per cent.

Anthracite coal consists almost entirely of free carbon and non-combustible. From eight specimens of American anthracite analyzed by Prof. Johnson, the mean composition is:

Carbon	86.76 per cent.
Volatile matter	4.98 " "
Moisture	1.18 " "
Non-combustible	6.97 " "
Sulphur	.11 " "

Bituminous coal consists of free carbon, hydrogen, oxygen, nitrogen, sulphur, and mineral compounds constituting the non-combustible matter. From twelve analyses of free burning bituminous coal Prof. Johnson obtains the following means:

Cumberland coal—

Carbon	73.72 per cent.
Volatile matter	14.20 " "
Sulphur	.12 " "
Moisture	1.56 " "
Non-combustible	10.40 " "

Pennsylvania coals—

Carbon	72.00 per cent.
Volatile matter	16.01 " "
Sulphur	.72 " "
Moisture	1.14 " "
Non-combustible	10.13 " "

Prof. Johnson's analyses of eleven varieties of Virginia caking bituminous coals furnishes as a mean—

Carbon	58.01 per cent.
Volatile matter	29.23 " "
Sulphur	.90 " "
Moisture	1.36 " "
Non-combustible	10.50 " "

Pittsburg coal (known in the market as Youghiogheny), consists of—

Carbon	54.93 per cent.
Volatile matter	36.60 " "
Moisture	1.40 " "
Non-combustible	7.07 " "

Newcastle (England) coal has the following composition—

Carbon	56.99	per cent.
Volatile matter	35.59	" "
Sulphur	.23	" "
Moisture	1.79	" "
Non-combustible	5.40	" "

The following is an analysis of Pittsburg coal, No. 2, by Prof. Bruno Kniffler, Cincinnati, 1879—

Fixed carbon	61.038	per cent.
Volatile matter	32.750	" "
Sulphur	0.863	" "
Moisture	2.307	" "
Ash	3.042	" "

Of fifty analyses of Indiana coals the following is a mean—

Carbon	51.20	per cent.
Volatile matter	42.79	" "
Non-combustible	6.01	" "

The following composition of Ohio coals is obtained from the "Geology of Ohio," volume II., being a mean of fifty-seven analyses, chiefly by Prof. Wormley—

Carbon	56.62	per cent.
Volatile matter	35.03	" "
Moisture	3.19	" "
Non-combustible	5.16	" "

Coke is the product of coal after eliminating the volatile matter, The process is conducted either in retorts, as gas coke, or in coke ovens. The latter is preferable for furnace fuel. Coke contains, as a mean—

Carbon	85.00	per cent.
Non-combustible	15.00	" "

Wood consists of—

Carbon	50.00	per cent.
Oxygen	42.00	" "
Hydrogen	5.25	" "
Non-combustible	2.75	" "

The oxygen and hydrogen exist in proportions to form water, and the carbon alone is useful in giving out heat. For equal weights the calorific power of all woods used for fuel is the same. Exceptions should be made of woods of the same family as the fir and pine, as these contain a small quantity of turpentine, which is a hydro-carbon.

Peat, or vegatable fuel, consists of—

Carbon	58.00	per cent.
Hydrogen	6.00	" "
Oxygen	31.00	" "
Non-combustible	5.00	" "

Lignite, although not generally classed as a separate fuel, occupies a position between peat and fully developed bituminous coal. Its composition is, as a mean—

Carbon	39.00 per cent.
Oxygen	10.00 " "
Hydrogen	2.50 " "
Non-combustible	48.50 " "

The fact is established by geological investigation, that anthracite and bituminous coals, and lignite are of vegetable origin. Thus, wood consists chiefly of carbon, hydrogen, and oxygen. By a process of natural evolution the wood suffers a loss of each of these elements, but principally hydrogen and oxygen, when we have lignite. This sustains a further loss of nearly all its oxygen, more than half its hydrogen, and a large percentage of carbon, when bituminous coal is the result. This suffers a further loss of a small percentage of carbon, and nearly all its hydrogen and oxygen, and anthracite coal is the result. This, finally, suffers a loss of all its oxygen, nearly all its hydrogen, and nearly pure carbon or graphite is the result.

The following table of composition of combustibles is from analyses by Peclet and others:

Elements.	Coal.	Coke.	Wood.			Peat.	
			Perfectly Dry.	Ordinary State.	Charcoal.	Perfectly Dry.	Ordinary State.
Carbon	.812	.850	.510	.408	.930	.580	.464
Hydrogen	.048	..	.053	.042	..	.060	.048
Oxygen	.054	..	.417	.334	..	.310	.248
Nitrogen and Sulphur	.031	..	..	..	..	..	.
Water	..	.		.200	..	..	.200
Ashes	.055	.150	.020	.016	.070	.050	.040
Total	1.000	1.000	1.000	1.000	1.000	1.000	1.000

Elements.	Naphtha.	Oil of Turpentine.	Alcohol.	Olive Oil.	Sulphuric Ether.	Tallow.	Beeswax.
Carbon	.850	.884	.5198	.7721	.6531	.790	.816
Hydrogen	.150	.116	.1370	.1336	.1333	.117	.139
Oxygen	..	..	.3432	.0943	.2136	.093	.045
Total	1.000	1.000	1.000	1.000	1.000	1.000	1.000

The following data is taken from the author's report to A. A Freeman & Co., New York, upon experiments at their flouring mill, La Crosse, Wis., with coal, pine slabs, and hard wood for steam purposes:

Fuel	Coal.	Pine Slabs.	Hardwood.
Date of trial	M'ch 13.	Mch. 14.	Mch. 14.
Duration of trial, hours	10	5	5
Mean pressure, by boiler gauge, pounds	92.876	93.325	90.10
Mean temperature of feed to boilers, Fahr	114.324	109.22	113
Total water pumped into boilers, pounds	50371.28	24608	29574.16
Water entrained in the steam, pounds	6467.7	3159.66	3797.32
Net steam furnished, pounds	43903.58	21448.34	25776.84
Total fuel burned, pounds	5350	6995	9100
Steam per pound of coal from feed, pounds	8.206	3.066	2.832
Steam per pound of coal from and at 212, pounds	9.639	3 617	3 324
Relative efficiency	100	37.52	34.48
Cost, coal per ton, slabs and hardwood per cord, dollars	4.50	1.25	3.00
Relative cost for equal effects	122.86	100	131.43

PRACTICAL RESULTS WITH DIFFERENT COALS.

The following extracts, from reports by the author upon test trials of various fuels under various conditions will be of interest as showing the results of practice. Of course it will not be assumed that the higher economies are due alone to the excellence of the fuel, nor that the low economies are due to lack of quality in the fuel. The skill of the fireman usually plays such an important part in the manipulation of a combustible, that these comparisons must be accepted only as approximative.

Massillon (Ohio), Coal—Bituminous.

Milwaukee Milling Co., March, 1879.

Number of boilers	2
Kind of boilers	Tubular.
Heating surface, square feet	1605 126
Ratio; heating to grate surface	34.80
Hours of trial	10
Average steam pressure, pounds	88 89
Average temperature of feed water	83.238
Total (net) steam pounds	38 390
Total coal burned, pounds	5 050
Steam, per pound of coal from and at 212 Fahr., pounds	8.905
Percentage of non-combustible	6.87

BRIAR HILL COAL (OHIO).

Germain & Co.'s Elevator, Milwaukee, March, 1879.

Number of boilers	1
Kind of boilers	Tubular.
Heating surface, square feet	325.95
Ratio; heating to grate surface	32.595
Hours of trial	7
Average steam pressure, pounds	85 96
Average temperature of feed water	195 643
Total (net) steam pounds	2427.811
Total coal burned, pounds	227
Steam, per pound of coal from and at 212 Fahr., pounds	11.416
Percentage of non-combustible	5.24

WILMINGTON COAL (ILLINOIS).

A. A. Freeman & Co., La Crosse, Wisconsin, March, 1879.

Number of boilers	2
Kind of boilers	Tubular.
Heating surface, square feet	1536 92
Ratio; heating to grate surface	29.70
Hours of trial	10
Average steam pressure, pounds	92 876
Average temperature of feed water	114 324
Total (net) steam pounds	43903.58
Total coal burned, pounds	5350
Steam, per pound of coal from and at 212 Fahr., pounds	9.639
Percentage of non-combustible	7.30

PITTSBURGH COAL (PENNSYLVANIA).

Hunt Street Pumping Station, Cincinnati, June, 1879.

Number of boilers	2
Kind of boilers	6 flue.
Heating surface, square feet	1082 98
Ratio; heating to grate surface	56.86
Hours of trial	86
Average steam pressure, pounds	128 00
Average temperature of feed water	215 26
Total (net) steam pounds	147109.00
Total coal burned, pounds	14100 00
Steam, per pound of coal from and at 212 Fahr., pounds	10 806
Percentage of non-combustible	3.06

PITTSBURGH COAL.

Millcreek Distilling Co., Cincinnati, September, 1882.

Number of boilers	2
Kind of boilers	Babcock and Wilcox, Sectional.
Heating surface, square feet	2640
Ratio; heating to grate surface	60.352
Hours of trial	10
Average steam pressure, pounds	64 09
Average temperature of feed water	136.15
Total (net) steam pounds	106728
Total coal burned, pounds	1200
Steam, per pound of coal from and at 212 Fahr., pounds	9.57
Percentage of non-combustible	4 890

Erie Coal.

N. K. Fairbank & Co., Chicago, June, 1882.

Number of boilers	1
Kind of boilers	Tubular.
Heating surface, square feet	758.173
Ratio; heating to grate surface	42.12
Hours of trial	9
Average steam pressure, pounds	41.882
Average temperature of feed water	173.870
Total (net) steam pounds	22673.834
Total coal burned, pounds	2914
Steam, per pound of coal from and at 212 Fahr., pounds	8.282
Percentage of non-combustible	4.890

Lehigh Coal (Pennsylvania).

Evansville Pumping Station, Evansville, Indiana, January, 1881.

Number of boilers	2
Kind of boilers	12-flue.
Heating surface, square feet	932.018
Ratio; heating to grate surface	20.7115
Hours of trial	24
Average steam pressure, pounds	95 427
Average temperatnre of feed water	121.917
Total (net) steam pounds	64949.514
Total coal burned, pounds	8916
Steam, per pound of coal from and at 212 Fahr., pounds	8.251
Percentage of non-combustible	11.47

Lackawanna Coal (Pennsylvania).

Peoria Pumping Station, Peoria, Illinois, March, 1882.

Number of boilers	2
Kind of boilers	Tubular.
Heating surface, square feet	1955.0048
Ratio; heating to grate surface	44.43
Hours of trial	18
Average steam pressure, pounds	79.076
Average temperature of feed water	118 71
Total (net) steam pounds	53986.214
Total coal burned, pounds	8900
Steam, per pound of coal from and at 212 Fahr., pounds	6.87
Percentage of non-cumbustible	16.326

Lackawanna Coal.

Saratoga Pumping Station. Saratoga, N. Y., November, 1882.

Number of boilers	2
Kind of boilers	Tubular.
Heating surface, square feet	2957.5
Ratio; heating to grate surface	51.89
Hours of trial	20
Average steam pressure, pounds	76.644
Average temperature of feed water	169.175
Total (net) steam pounds	70582.779
Total coal burned, pounds	6750
Steam, per pound of coal from and at 212 Fahr., pounds	11.286
Percentage of non-combustible	3.2

KANAWHA "SLACK" AND COKE "BREEZE.'

Cincinnati Gas Works, November, 1882.

	Breeze.	*Slack.*
Number of boilers	3	3
Kind of boilers	Locomotive fire-box.	Locomotive fire-box.
Heating surface, square feet	1768.958	1768.958
Ratio; heating to grate surface	31.799	31.799
Hours of trial	10	10
Average steam pressure, pounds	59.35	62.573
Average temperature of feed water	147.93	150.58
Total (net) steam pounds	56673.226	58777.928
Total coal burned, pounds	10348	9922
Steam, per pound of fuel from and at 212 Fahr., pounds	6.006	6.486
Percentage of non-combustible	13.08	8.97

HIGHLAND BLOCK COAL (INDIANA).

Gibson & Co. Flour Mill, Indianapolis, August, 1877.

Number of boilers	2
Kind of boilers	6-flue.
Heating surface, square feet	931.68
Ratio; heating to grate surface	24.52
Hours of trial	8
Average steam pressure, pounds	81.37
Average temperature of feed water	195
Total (net) steam pounds	23542.4
Total coal burned, pounds	4814
Steam, per pound of coal from and at 212 Fahr., pounds	5.24
Percentage of non-cumbustible	Not measured.

HEAT.

The fact that heat possesses energy, and that energy being ponderable, has, up to a very recent period, induced the belief that heat was a material substance. It is now well known, however, that heat is a state of matter, and that while it is referable to cause and effect, and its force, like gravity, governed by established laws, it is determinable as a condition of matter, and possesses no independent existence. In 1798, Count Rumford published a memoir of his experiments on the production of heat by friction. Up to this time the theory of material substance prevailed. Heat was supposed to be a fluid, and, like air and water, capable of uniting with other substances according to their several capacities for heat.

As proof that heat was simply a condition of matter, Sir Humphrey

Davy reduced a block of ice to liquid water by friction alone. Thus by the expense of a certain energy he developed heat sufficient to melt the ice. If heat was matter, this would have been impossible, since matter can not be created.

The experiments of Prof. Tyndall have done more to increase our knowledge of the laws and phenomena of heat than that of any other scientist.

The mechanical equivalent of heat as determined by Mr. Joule, of Manchester, is one of the most useful factors in heat investigation. This gentleman, by very careful and precise experiments, extending through several years, established the value in foot pounds of work of a British thermal unit, and conversely the energy requisite to produce a unit of heat. Mr. Joule determined the energy required to add one thermal unit to a pound of water to be 772 foot pounds, and this value is usually represented by the letter "*J*" in heat formulæ.

The temperature corresponding to the disappearance of gaseous elasticity is termed the absolute zero; and this point has been determined in accordance with the Guy Lussac law, as modified by the later experiments of Rudberg, Magnus, and Regnault. Guy Lussac's experiments have shown *that for the same density the tensions and for the same tensions the volume of one and the same quantity of air increases with the temperature.* Experiment has shown the co-efficient of expansion of air to be .0020276 on Fahrenheit's scale, hence absolute zero $= \frac{1}{.0020276} = 493.20$ below the temperature of melting ice, or $493.2 - 32 = 461.2$ below Fahr. zero. Thus to know the absolute temperature at any point above Fahr. zero, add 461.20. Example—Observed temperature 60°; absolute temperature 521.20.

Specific heat is the capacity of a body to absorb heat, as compared with water. Water possesses the highest specific heat of any known substance except hydrogen gas. Thus while *one* thermal unit will elevate the temperature of one pound of water *one* degree at 60° Fahr., and pressure of one atmosphere, 3.4046 thermal units are requisite to elevate the temperature of the same weight of hydrogen *one* degree under same pressure and temperature.

If the Mariotte law were strictly correct, the specific heat of gases would be the same for constant volume or constant pressure; but Regnault's experiments have shown that the specific heat is greatest for constant pressure.

Thermometers are instruments to measure variations of temperature. For ordinary use the mercurial thermometer is sufficient, but for scientific research the air thermometer is employed. For temperatures below the point of congelation of mercury (—38° Fahr.) spirit

thermometers are used. In Europe, except Great Britain, Spain, and Holland, the Centigrade scale is used. In Great Britain, Holland, and the United States, Fahrenheit's scale is used. In Spain, Reaumur's scale is used. In the Centigrade scale the zero is taken at temperature of melting ice, while the boiling point of water under pressure of one atmosphere is taken at 100°. On the Fahrenheit scale the zero point is taken at 32° below the temperature of melting ice, and the boiling point at pressure of one atmosphere becomes 212°. By comparison, 180° of Fahrenheit scale equals 100° of Centigrade scale. Hence, to reduce a reading on Centigrade scale to corresponding temperature on Fahrenheit scale,

$$\frac{C \times 9}{5} + 32 = F,$$

BOILING POINTS OF LIQUIDS UNDER PRESSURE OF ONE ATMOSPHERE.

Substance.	Temp. Fahr.
Sulphuric ether	100
Sulphuret of carbon	118.4
Ammonia	140
Chloroform	140
Bromine	145
Wood spirits	150
Alcohol	173
Benzine	176
Water	212
Sea water	213.2
Saturated brine	226
Nitric acid	248
Oil of turpentine	815
Phosphorus	554
Sulphur	570
Sulphuric acid	590
Linseed oil	597
Mercury	648

TEMPERATURE OF FIRE AS INDICATED BY COLOR.

The following table may be used for approximating temperature at a glance; where accuracy is required, calorimeter tests should be resorted to for temperature.

		Pouillet.
Faint red	indicates about	960 Fahr.
Dull "	" "	1290 "
Brilliant red	" "	1470 "
Cherry "	" "	1650 "
Bright cherry red	" "	1830 "
Dull orange	" "	2010 "
Bright orange	" "	2190 "
White heat	" "	2370 "
Bright white	" "	2550 "
Brilliant white	" "	2730 "

TEMPERATURE BY CALORIMETER.

Calorimeter tests for temperatures below the melting point of wrought iron are made in the following manner: A small bar of iron weighing one or two pounds is suspended in a flue or in a fire box, as the case may be, and is allowed to take the temperature of the surrounding hot gas. The time required in any particular case should be determined by experiment. Suppose three bars of similar weight and similarly disposed in a flue or fire box, are allowed to remain two and one-half minutes, five minutes, and ten minutes respectively, meanwhile the conditions of fire are not materially changed. Then, if the resulting temperatures are substantially alike, the shorter period of time is sufficient to acquire the full temperature of hot gas; if the two longer period bars are alike in temperature, then five minutes is known to be a sufficient length of time to acquire the full temperature of hot gas. If the ten minute bar shows the greatest temperature then further tests with ten minutes as a mean are required.

In making a preliminary test, the ten minute bar should first be introduced, and five minutes later the five minute bar introduced, and two and one-half minutes later the two and one-half minute bar should be introduced. In other words, the bars should all *leave* the flue or fire box at the same time.

The time required to heat the bars to the full temperature of the hot gas, is in an inverse ratio to the temperature of the gas. Thus, if five minutes be sufficient to acquire a temperature of 2500 F. considderably more time will be required to assume a temperature of 500 F.

After determining the time required to acquire the temperature, the operation consists simply in cooling down the bars (respectively) in a known weight of water, noting the temperature of the water before the bar is dropped into it, and after the bar and water have assumed a like temperature. Several bars are used only, that the results of any one test may be more reliable.

To illustrate the method:

Let w = weight of bar when it enters the water; W = weight of water heated; T = initial temperature of water, and T_1 = final temperature of water and iron; S the specific heat of water at temperature T, S_1 the specific heat of water at temperature T_1, and St the specific heat of iron, which may be taken at .1138 for normal temperatures. Then range $R = T_1 . S_1 - T . S$ and heat units added to water per pound of iron $H = \frac{W\,R}{w}$ and temperature of iron

EUREKA FURNACE ATTACHMENT.

	Thermal units.	Steam.	Per cent.
Steam	8384.555	8.679	54.094
Chimney gas	2616.616	2.709	16.881
Vapor of water in air	75.697	.078	.488
Moisture in coal	29.092	.030	.187
Combustible gas	620.000	.642	4.000
Radiation	3774.040	3.907	24.350
	15500.000	16.045	100.000

PRICE FURNACE.

	Thermal units.	Steam.	Per cent.
Steam	12025.690	12.449	77.538
Chimney gas	1772.842	1.835	11.437
Vapor of water in air	60.390	.062	.389
Moisture in coal	28.874	.030	.186
Combustible gas	387.500	.401	2.500
Radiation	1224.704	1.268	7.905
	15500.000	16.045	100.000

MURPHY FURNACE.

	Thermal units.	Steam.	Per cent.
Steam	12487.920	12.928	80.567
Chimney gas	1033.118	1.069	6.665
Vapor of water in air	32.103	.083	.207
Moisture in coal	27.086	.028	.174
Combustible gas	387.500	.401	2.500
Radiation	1532.273	1.586	9.887
	15500.000	16.045	100.000

The distribution of the heat in the several furnaces has been calculated in the following manner:

HEAT IN STEAM.

Let S represent the steam furnished per pound of coal from and at 212 Fahr., and c the combustible in decimal of the net coal charged; then, $\frac{S}{c} = S' =$ the steam furnished per pound of combustible.

Each pound of steam from and at 212 F. contains 966 thermal units, and $966\ S' = T =$ thermal units found in the steam per pound of combustible, and $\frac{966\ S'}{15500} = K =$ decimal of total heat found in the steam.

HEAT IN CHIMNEY GAS.

Let T be the temperature of the gas in front connection, and t the temperature of external air. Let A equal the weight of hot gas per pound of combustible. The mean specific heat of the gas is probably .238; then $A\ (T - t)\ .238 = H =$ thermal units accounted for per pound of combustible in the hot gas; and $\frac{H}{966} = S' =$ steam from and at 212 Fahrenheit, represented by the heat resident in the hot gas as it entered the chimney; and $\frac{H}{15500} = K =$ decimal of the total heat found in the waste gases.

The weight (34.7898 pounds) of air per pound of combustible, charged to the Fisher Furnace, does not include the air that entered the furnace through and behind the bridge wall. From the area of openings through and behind the bridge wall, it is estimated that the weight of air thus conducted into the furnace was equal to the quantity required to support combustion, whence the weight of hot gas passing up the chimney becomes—(weight of air entering fire chamber $\times$ 2) + 1.

HEAT IN VAPOR OF WATER.

Let g be the weight in grains of the vapor of water in a cubic foot of air at maximum saturation, as shown by temperature of deposition on the hygrometer, and C the correction for the absolute dryness observed, according to Mr. Foggo: then $\frac{g}{C} = g' =$ the weight in grains of the vapor of water per cubic foot of air supplied to the furnace. Let W be the weight per cubic foot of water at temperature of air, and 815 the ratio of the weight of water to air at same temperature and pressure; then $\frac{W}{815} = W' =$ the weight of a cubic foot of air, and $\frac{1}{W} = V =$ cubic feet of air per pound; then $\frac{g' V}{7000} = D =$ weight in decimal of pound of the vapor of water per pound of air supplied to the furnace.

The values of the vapor of water per pound of air supplied, in the data from the trials, were calculated in accordance with these formulæ.

Let A, as before, be the weight of hot gas per pound of combustible passing up the chimney; then $A - 1 = A' =$ the weight of hot gas due the air. Let T be the temperature of hot gas, and t the temperature of external air. The mean specific heat of the vapor is probably .4805;

then $A', D(T-t)\ .4805 = H =$ thermal units per pound of combustible absorbed by the vapor of water in the air, and $\frac{H'}{15500} = K =$ decimal of the total heat found in the vapor of water; and finally $\frac{H}{966} = S' =$ steam from and at 212 Fahrenheit represented by the heat in the vapor of water.

MOISTURE IN COAL.

Let G equal the weight in decimal of a pound of the moisture in the coal; and c the decimal of the combustible for the respective trials; then $\frac{G}{c} = G' =$ the weight of the moisture per pound of combustible. The pressure of vaporization would be that of the atmosphere corresponding to a temperature T', and total heat L.

Let T, as before, be the temperature of waste gases entering the chimney, and t the temperature of external air; then $G'\ (T - T')\ .4805 = H' =$ thermal units per pound of combustible represented by the super heat in the moisture, and $G'\ (L - t) = H^2 =$ thermal units per pound of combustible represented by the saturated vapor; then $H' + H^2 = H =$ thermal units per pound of combustible found in the moisture from the coal, and $\frac{H}{15500} = K =$ decimal of total heat found in the moisture; and, finally $\frac{H}{966} = S' =$ steam from and at 212 Fahrenheit due the heat taken up by the moisture.

COMBUSTIBLE GAS AND RADIATION.

The combustible gas has been approximated upon the known condition of fire: and the heat lost by *conduction* and *radiation* has been taken as the difference between the heat actually accounted for and the total heat per pound of combustible. The heat lost by conduction, radiation, and by contact of air, may be estimated by the formule previously given when it is desired to know each separately.

In determining the efficiency of a steam boiler furnace, it is sufficient to know the value of the fuel, and the co-efficients of this value represented by the steam and chimney gas; the balance may be charged to conduction, radiation, and loss of heat by contact of air, without sensible error, as the heat latent in the gases of combustion, and that absorbed by moisture of air and coal, are usually too insignificant to be worthy of special consideration.

COEFFICIENTS OF EXPANSION OF BODIES BY HEAT.

Being the increase in length for each degree of Fahrenheit scale from 32 to 212.

Fire-brick	.000002349
Marble, black	.000002407
" white	.000003633
Granite	.000004386
Glass, tube	.000004567
" plate	.000004769
White pine	.000002556
Platina	.000004835
Slate	.000005764
Cast iron	.000006167
Wrought iron	.000006689
Steel	.000006614
Copper	.000010088
Brass, cast	.000010417
" plate	.000010450
" wire	.000010723
Antimony	.000006006
Bismuth	.000007716
Gold	.000008122
Sandstone	.000009680
Silver	.000011121
Tin	.000013102
Lead	.000015876
Zinc	.000017268
Pewter	.000012685

EXPANSION BY VOLUMES.—(Box.)

For one degree temperature Fahr.

Mercury	.00010054
" glass tubes	00008684
Alcohol	.0006318
Linseed oil	.0004030

LOSS OF HEAT BY CONDUCTION.

The amount of heat lost by conduction through a plate or wall (as the wall of a steam boiler furnace) depends upon the difference of temperature of the two surfaces, or upon the difference of temperature of the air or other matter within and of the air or other matter without, upon the thickness of the wall or plate, and upon the conducting power of the material.

The following table, from Peclet's experiments, indicates the units of heat per hour, per square foot of surface, transmitted through a plate or wall of one (1) inch thickness:

CONDUCTING POWER OF MATERIALS = C.

Copper	$C =$	515.
Iron	"	233.
Zinc	"	225.
Lead	"	113.
Marble, grey, fine-grained	"	28.
" white, coarse-grained	"	22 4
Stone, calcareous, fine	"	16 7
" " ordinary	"	13 68
Glass	"	6 6
Brick Work	"	4 83
Plaster	"	3 86
Oak, perpendicular to fibers	"	1.70
Walnut, " "	"	.83
Fir, " "	"	.748
" parallel to fibers	"	1.37
Walnut, parallel to fibers	"	1 40
Gutta Percha	"	1 38
India Rubber	"	1 37
Brick Dust, fine	"	1.33
Coke, fine	"	1.29
Cork	"	1.15
Chalk, powder	"	.869
Charcoal, powder	"	.639
Straw, chopped	"	.563
Coal, small, sifted	"	.547
Wood Ashes	"	.531
Mahogany Dust	"	.523
Canvas of Hemp, new	"	.418
Calico, new	"	.402
White Writing Paper	"	.346
Wool, Cotton, or Sheep	"	.323
Eider Down	"	.314
Blotting Paper	"	.274

Let T = temperature of the hotter surface of a wall or plate and T' = temperature of opposite surface, t = thickness of same in inches, S = area in square feet, and C = the conducting power of the material. Then

$$H = \frac{C(T - T')S}{t}$$

When H = heat units transmitted by conduction per hour.

Suppose the cover of a steam chest is of an average 1.25 inches thick, of cast iron, 15 inches wide $\times$ 24 inches long = 2.5 square feet, and the temperature of the steam and sensibly of the plate within is 320 F., and of the plate without 290 F., then loss of heat by conduction

$$H = \frac{233 \times (320 - 290) \times 2.5}{1.25} = 13980$$ units per hour, *equivalent to one pound of good coal.*

Suppose this cover was well lagged with walnut with grain of wood parallel to plane of cover, of staves one inch thick, then the loss of heat would be represented solely by the conducting power of the lag-

ging, and assuming inner surface of lagging to have a temperature of 310 and outer surface to have a temperature of 120; then loss of heat becomes $H = \frac{.83\,(310 - 120)\,2.5}{1} = 394.25$ units per hour, or $\frac{394.25}{13980} = .0282$ $= 2.82$ per cent of the loss by naked plate.

LOSS OF HEAT BY CONTACT OF AIR.

The loss of heat by contact with air for a vertical plane, according to Box, varies inversely as a certain function of the $\sqrt{\ }$ of the head (H') or by formula $A = .361 + \frac{.233}{\sqrt{10}}$ where A = loss in units of heat per hour, per square foot of surface, for a difference of one degree Fahrenheit.

Suppose a wall 10 feet high, 20 feet long, the surface temperature of which is 100 F., and the air in contact with it 75 F., what will be the loss by contact of air per square foot of surface.

Then $A = .361 + \frac{.233}{\sqrt{10}} = .4347$ unit, and loss for entire surface, $H' = .4347 \times 10 \times 20 = 86.94$ units. Suppose the height is 25 feet, instead of 10 feet, then loss per square foot of surface per hour would be $A = .361 + \frac{233}{\sqrt{25}} = .4076$ unit, and for entire wall, $H' = .4076 \times 25 \times 20$ $= 203.8$ units.

The loss of heat per square foot of surface, per hour, for a difference of one (1) degree temperature, for a horizontal cylinder, is expressed by the formula.:

$A' = .421 \frac{.307}{r}$ where A' = the loss in heat units per square foot of surface per hour, for a difference of one degree temperature, and r = radius of cylinder or arc in inches. (This formula only takes cognizance of the loss by the convex or concave surface, and does not account for loss at the ends.)

Suppose a convex surface, the length of arc of which is 2 feet, axial length 5 feet, and radius of arc 24 inches, what will be the loss of heat by contact of air, per square foot of surface, for one (1) degree difference of temperature?

$A' = .421 \frac{.307}{24} = .43379$ unit and loss for entire surface.

$H' = .43379 \times 2 \times 5 = 4.3379$ units per hour.

The loss of heat for a sphere by contact of air, for a difference

of temperature of one degree per square foot of surface, per hour, is expressed by the formula:

$A'' = .3634 + \frac{1.0476}{r}$ where A'' = units of heat per hour per square foot of surface, and r = radius of sphere.

Suppose a sphere 3 feet in diameter. at a temperature of 80 F., whilst temperature of surrounding air is 79 F., what will be the loss of heat per hour per square foot of surface?

$A'' = .3634 + \frac{1.0476}{18} = .4216$ unit, and for entire surface of sphere

$H''' = .4216 \times (3.416 \times 3^2) = 11.91$ units per hour.

The loss of heat by contact of air is independent of the material, and dependent only upon the difference of temperature and form of the surface.

Thus, for same area and form of surfaces, and same differences of temperature, copper, iron, wood, or brickwork would lose heat at the same rates per square foot of surface per hour.

LOSS OF HEAT BY RADIATION.

The loss of heat by radiation is practically independent of space radiated through, and dependent only upon the radiating power of the substance, the radiating surface, and difference of temperature of the radiating and recipient surfaces.

The following table of radiating values from Peclet represents the loss or gain of heat per square foot of surface per hour in heat units, for one (1) degree F. difference of temperature:

Polished silver	R =	.02657
Copper	"	.03270
Tin	"	.04395
Brass or zinc polished	"	.04906
Tinned iron	"	.08585
Sheet iron	"	.09200
Lead	"	.13286
Ordinary iron	"	.56620
Glass	"	5948
Cast iron, new	"	.6480
Chalk	"	.6786
Sheet or cast iron (corroded)	"	.6868
Wood saw-dust, fine	"	.7215
Building stone, wood, brickwork	"	.7358
Sand, fine	"	.7400
Calico	"	.7461
Woolen stuffs, any color	"	.7522
Silk stuffs, oil paint	"	.7583
Paper, any color	"	.7706
Lampblack	"	.8196
Water	"	1 0853
Oils	"	1.4800

The radiating and absorbing powers of the same body are equal

MELTING POINTS OF SOLIDS.

From Rankine and Pouillet.

Cast iron	Rankine.	3479	Fahr.
" " very fusible.	Pouillet.	2010	"
" " white maximum.	"	2010	"
" " second melting	"	2190	"
Gold	Rankine.	2590	"
" very pure	Pouillet.	2280	"
" standard coin.	"	2156	"
Copper	Rankine.	2548	"
Silver	"	1280	"
" very pure.	Pouillet.	1830	"
Brass	Rankine.	1869	"
"	Pouillet.	1650	"
Antimony	"	810	"
Zinc.	Rankine.	700	"
"	Pouillet.	793	"
Lead.	"	630	"
Bismuth	Rankine.	493	"
"	Pouillet.	518	"
Tin	Rankine.	426	"
"	Pouillet.	455	"
Sulphur	Rankine.	228	"
"	Pouillet.	239	"
Wax, white.	"	154	"
" unbleached	"	143	"
Spermaceti	"	120	"
Stearine	"	109	"
"	"	120	"
Phosphorus	"	109	"
Tallow	"	92	"
Oil of Turpentine	"	14	"
Mercury	Rankine.	—38	"
"	Pouillet.	—40	"
Common salt 1, water 3	Ure.	4	"
Sulphuric acid, sp. gr. 1.6415		45	"
" ether		46	"
Nitrate of potash (saltpetre)		630	"

MELTING POINT OF ALLOYS.

(Tin, Lead, and Bismuth.) *Rankine and Pouillet.*

Tin 1,	Lead 3.			Pouillet.	504	Fahr.
" 1,	" 1.			"	466	"
" 2,	" 1.			"	385	"
" 3,	" 1.			"	367	"
" 3,	" 2.			Rankine.	334	"
" 4,	" 1.			Pouillet.	372	"
" 5,	" 1.			"	381	"
" 2,	" 0,	Bismuth 1		Rankine.	334	"
" 1,	" 0,	" 1		"	286	"
" 1,	" 1,	" 4		Pouillet.	201	"
" 3,	" 5,	" 8		"	212	"
" 3,	" 5,	" 8		Rankine.	210	"
" 3,	" 2,	" 5		Poulilet.	212	"
" 4,	" 1,	" 5		"	246	"
" 3,	" 0,	" 1		"	392	"

FRICTION.

Friction is assumed to be independent of surfaces in contact, and directly as the force with which bodies are pressed together. Friction has been supposed to be independent of velocity, but the experiments of Bochet in 1858, seem to show that friction diminishes as the velocity increases. Weisbach, however, suggests that Bochet's deductions require further proof before they can be accepted as conclusive.

Adhesion is sometimes confounded with friction, but the laws of adhesion are the opposite of those for friction. Adhesion varies directly as the surfaces in contact, and is independent of the force with which bodies are pressed together; while for friction the reverse is true. With large surfaces and small pressures, the adhesion is great as compared with the friction.

The co-efficicent of friction varies with different materials, and with different conditions of the same material. Friction also largely depends upon the lubricant, and the manner in which it is supplied to the surfaces in contact. Continuous lubrication is to be preferred, and the supply should be carefully adjusted to the condition of the surfaces.

The angle of friction, or repose, is the greatest angle at which one body will rest upon another without sliding off. The tangent of this angle is the co-efficient of friction.

Friction is known either as sliding friction or rolling friction. Sliding friction is developed in the motion of a cross-head in the guides of an engine. Rolling friction is developed when a locomotive draws a loaded train.

From Morin's experiments on the friction of journals revolving on cast iron, or bronze bearings, it appears that the co-efficient for continuous labrication is .054, and with intermittent lubrication .07 to .08.

Hirn made the friction of journals in their bearings the subject of experiments, from which he deduces the following results:

"The mediate friction (surfaces lubricated) is dependent not only upon the pressure and the nature and character of the rubbing surfaces and of the unguent, but also upon the velocity and the temperature of the rubbing surfaces, as well as upon the magnitude of the surfaces.

"The friction is directly proportional to the velocity when the temperature is constant; and if the temperature is disregarded, it increases with the square root of the velocity.

"The mediate friction is also proportional to the square root of the rubbing surfaces, as well as the square root of the pressure."

According to Weisbach, the friction of revolving journals increases with the radius and number of revolutions. Thus the moment of friction depends upon the force with which surfaces are pressed together, but the friction (work) is the moment of friction into the space described. Hence $F = W.\ K.\ S.$ = friction per revolution, when W = the weight in bearing, K = co-efficient of friction, and S = the circumfrence of journal.

A fly wheel weighs 16,000 pounds; the shaft (10″ diameter, 8′ long) weighs 2130.5 pounds; the crank weighs 469.5 pounds. Suppose the center of gravity of the mass to be in a plane midway between the centers of the journals, then the friction, per formula, with continuous lubrication of surfaces becomes $\frac{18600}{2} \times .054 = 502.2$ moment of friction, and the work of friction per revolution for each of two journals,

$$\frac{10. \times 3.1416}{12} \times 502.2 = 1314.76 \text{ ft. pounds.}$$

The bearing is 20″ long, and the pressure per horizontal inch becomes $\frac{9300}{200} = 46.50$ pounds.

FRICTION OF SLIDE VALVE.

The expenditure of power in moving the ordinary slide valve, is the moment of friction into the travel, and the moment of friction is a function of surfaces in contact, and the unbalanced load on the valve (the total load being the area of back of valve parallel to the plane of contact, into the pressure in the chest). From this it appears that the smaller the valve for a given effect, the less the power absorbed in moving it. An erroneous idea prevails among certain builders of engines that the friction of the valve is independent of its size, and only dependent upon the area of the steam passages which it covers. The following demonstration of the friction of slide valve by the author, is taken from the "Engineering and Mining Journal," of Feb. 3, 1877:

"Let A = area of valve parallel to face, impinged upon by the steam in the chest; and P = the intensity of pressure in the chest. If A were a constant for all positions of valve, then the total load perpendicular to the plane of motion becomes $A \times P$; and were it not that a portion of this quantity is neutralized in effect by a force also acting in a plane perpendicular to the face of valve and opposite to the force

$A\ P$, then $A\ P$, modified by a proper co-efficient, would represent the moment of friction at all points in the travel.

Let A' equal effective area of under side of valve referred to whole stroke of piston, and P' the corresponding mean pressure, then $A'\ P'$ is the neutralizing force; hence the moment of friction F is a function of $A\ P - A'\ P'$.

Let S be the travel of valve in feet and r the revolutions per second, then the expenditure of power in overcoming the friction of the valve is expressed by the equation,

$$H = \frac{F.\ S.\ r}{550} \times 2$$

Let H' be the indicated $H.\ P.$ of engine, then

$K = \frac{100\ H}{H'}$ = percentage of power expended in moving the valve.

The following data is from the author's experiments: Diameter of cylinder, 16″; piston speed, 400′; slide valve, 8.75″ × 14″; travel. 5″; area of steam ports, 15″; area of exhaust ports, 24″; width of steam port, 1.25″; exhaust, 2″; pressure in the chest, 85 pounds; steam cut off at ⅜ of piston stroke. The area of valve parallel to plane of contact is 122.5 square inch, and the total load 10,412 pounds.

From the diagram we have the following data: Mean pressure to cut-off, 57.44; from cut-off to release, 31.84; from release to end of stroke, 15.00 Return stroke, mean pressure, .75; from cushion to end of return stroke, 14.00.

The neutralizing effect during admission becomes

$$\frac{15 \times 1.25 \times 57.44.}{5} = 215.4 \text{ pounds.}$$

During expansion,

$$\frac{15 \times 1.25 \times 31.84}{5} = 119.4 \text{ pounds.}$$

During release,

$$\frac{15 \times .625 \times 15.00}{5} = 28.125 \text{ pounds.}$$

During exhaust (return) stroke (exhaust pocket in valve 12 × 3.75= 45 square inches),

$$45. \times .9 \times .75 = 30.375 \text{ pounds.}$$

During compression,

$$\frac{15 \times 1.25 \times 14.00}{5} = 52.5 \text{ pounds.}$$

Hence 10412 — 445.8 = 9966.2 pounds. With a co-efficient of friction

of .15 and revolutions per minute of 100, then the power absorbed by the valve becomes

$$\frac{9966.2 \times .15 \times 5 \times 100 \times 2}{33000 \times 12} = 3.77 \text{ H. P.}$$

The mean effective pressure by the diagram was 45.33 pounds, area of piston 201; hence indicated power of engine,

$$\frac{201 \times 400 \times 45.33}{33000} = 110.5 \text{ H. P.}$$

And percentum of power expended in moving the valve,

$$\frac{3.77 \times 100}{110.5} = 3.4.$$

The opinion entertained by certain engineers that the slide valve floats on a thin film of steam, is not only erroneous, but undesirable, for if the fit of the valve to its seat is such as to allow a circulation of steam, of maximum pressure sufficient to balance the load (in part), it is likewise sufficient to allow the passage of steam (between the valve face and seat) into the exhaust. Considering the intimate relation that must subsist between the valve and the seat, in order to *prevent* leakage into the exhaust, it is probable that the liquefaction of steam, due to the attraction of the metal surfaces, is sufficient to prevent the passage of steam under the valve.

ROLLING FRICTION.

Rolling friction increases with the pressure, and is inversely as the diameter of the rolling body.

The moving friction of locomotives is about 15 pounds per ton, and for cars from 6 to 11 pounds per ton.

Pivot friction is estimated as follows: Let R = radius in feet of pivot surface perpendicular to axis of rotation, K = the co-efficient of friction, and W = weight on pivot, then the friction F. per revolution becomes

$$F = \frac{R\ 6.2832 \times K \times W \times 2}{3}$$

Weight, 12,000 pounds; co-efficient of friction, .06; diam. of pivot flat bearing surface, 4″; desired friction per revolution,

$$F = \frac{2 \times 6.2832 \times .06 \times 12000 \times 2}{3 \times 12}\ 502.656$$

foot pounds.

CO-EFFICIENTS OF FRICTION.—(Weisbach.)

MATERIAL.	UNGUENT.	Co-eff. at rest.	Co-eff. in motion
Wood on wood, min.	Surfaces dry.	.30	.20
" " " mean	" "	.50	.36
" " " max.	" "	.70	.48
" " " min.	Water.	.65	
" " " mean	"	.68	.25
" " " max.	"	.71	
" " " mean	Hogs lard.	.21	.07
" " " min.	Tallow.	.14	.06
" " " mean	"	.19	.07
" " " max.	"	.25	.08
" " " min.	Polished & greasy.	.30	.08
" " " mean	" "	.35	.12
" " " max.	" "	.40	.15
" " metal	Dry Surfaces.	.60	.42
" " "	Water.	.65	.24
" " "	Hogs lard.	.12	.07
" " "	Tallow.	.12	.08
" " "	Polished & greasy.	.10	
Metal on metal, min.	Dry.	.15	.15
" " " mean	"	.18	.18
" " " max.	"	.24	.24
" " " min.	Olive oil.	.11	.06
" " " mean	" "	.12	.07
" " " max.	" "	.16	.08
" " " mean	Hogs lard.	.10	.09
" " " "	Tallow.	.11	.09
" " " "	Polished & greasy.	.10	
Thick sole leather on wood on edge	Dry.	43	.34
" " " " " flat	"	.62	.54
" " " " " on edge	Water.	.62	.31
" " " " " flat	"	.80	.36
" " " " " on edge	Olive oil.	.12	
" " " " " flat	" "	.13	
Stone on stone polished, min.	Dry.	.67	
" " " " max.	"	.75	
" " wrought iron, min.	"	.42	
" " " " max.	"	.49	
Hemp in ropes on wood, min.	"	.42	
" " " " " mean	"	.49	.45
" " " " " max.	"	.64	
" " " " " mean	Water.		.33
Bronze on lignum vitae (*Rankine.*)	"		.05
Smooth surfaces "	Random lubrica'n		.075
" " "	Continuous "		.050
" " best results "	" "		.033
Masonry on dry clay (*Trautwine.*)			.510
" " moist clay "			.33
" and brickwork dry "			.65
" " " wet mortar "			.47
" " " damp " "			.74
Brick on brick			.64
Leather belts on wood	Dry.	.47	
" " " metal	"	.54	

FRICTIONAL RESISTANCE OF WHEELED VEHICLES.

In pounds per ton (2240 pds.) of load.

(Omnibus with load 5758 pounds.) *D. K. Clark.*

	Pds. per ton.	Miles per hour.
Granite pavement	17.41	2.87
Asphalt "	27.14	3.56
Wood "	41.60	3.34
Good gravelly macadam. road	44.48	3.45
Granite macadam, road, new	101.09	3.51

Wagon weighing 2352 pounds, 2½ miles per hour.) *Macneil.*

Well made pavement	31.2	2.5
Road of broken stone on firm bed of large stones or concrete	44.	2.5
Road of thick coating of broken stone on earth	62.	2.5
Road of thick coating of gravel on earth	140.	2.5
Stage coach, 6 miles per hour	62.	
" " 8 " "	73.	
" " 10 " "	79.	

Mr. D. K. Clark proposes the following formula for resistance to traction in pounds per ton of stage coach in good metaled roads:

Let R = frictional resistance in pounds per ton of load.

v = speed of coach in miles per hour.

Then— $$R = 30 + 4v + \sqrt{10v}$$

Frictional resistance of horse cars about 26 pounds per ton of load.—(*Hughes.*)

Mr. D. K. Clark gives the following formulæ for resistance of trains on railways in ordinary practice:

Let R = resistance of train alone (2000 pounds) per ton, in pounds.

R' = resistance of engine and train per ton, (2000 pounds) in pounds, and

V = speed in miles per hour.

Then— $$R = \frac{\left(6 + \frac{V^2}{240}\right)1.5}{1.12} \text{ and } R' = \frac{\left(8 + \frac{V^2}{171}\right)1.5}{1.12}$$

From which it appears that of the total load, Mr. Clark allows more than *one-fourth* for resistance of engine alone. The authors experience with locomotives drawing freight trains, shows that *eighty-five* per cent of the total power developed is expended in moving the train, and only *fifteen* per cent absorbed by the the locomotive itself. (Vide Journal of the Franklin Institute, April and May, 1879.)

For speeds of 10 — 15 — 20 — 30 — 40 — 50 — 60 miles per hour.

The resistance of engine and train per ton (2000 pounds), is

11.51 2.48 13.85 17.76 23.25 30.29 38.91 pounds.

CHIMNEYS.

In estimating the stability of chimneys no attention is paid to the cohesive effect of the mortar joints. The weight alone is considered.

For wind pressure, the effective section of round and octagon chimneys, and chimneys of more than eight sides are taken by Rankine as one-half (.5) the actual diametrical section.

Thus, if a chimney has a mean diameter of 7 feet, and a height of 100 feet, the diametrical section would be $7 \times 100 = 700$ square feet of which $\frac{700}{2} = 350$ square feet is regarded as the effective surface acted upon by a gale of wind. And for square chimneys, the effective section is taken by Rankine as the actual diametrical section. From which it appears that a chimney the external mean section of which is 7 x 7 feet, and 100 feet high, presents twice the surface for wind pressure of an octagon or round chimney the mean diameter of which is 7 feet and of same height.

In designing a chimney it is desirable for a given cross section of flue and height of shaft, to have a minimum diametrical plane, and maximum load (within limits of safety) per square foot of base. It will be evident from the following equations that for a given height and weight of shaft, the stability increases with the rate of batter of outer surface

Professor Rankine gives the following formulæ for stability of chimneys:

Let H = height in feet from base of shaft or any given bed joint, to center of gravity of so much of chimney as lies above said base or bed joint.

W = weight in pounds of so much of shaft as lies above the base or any given bed joint.

q' = ratio of deviation of center of gravity of given section (shaft) of chimney, from true axis at base of section. (Thus, if a perpendicular let fall from center of gravity of section intersects diameter at base one inch from middle of diameter, and diameter at base is 10 feet, then $q' = \frac{1}{120} = .0083$.

In a chimney the axis of which is strictly vertical $q' = o$.

D = diameter of section of chimney (shaft) at base, in feet.

(Chimneys are generally built with a heavy plinth, the stability of which is relatively greater than of shaft, whence D is usually taken as at base of shaft.)

B = mean thickness of brick work above base of shaft, or above any given bed joint, in inches.

w = weight of brick work per cubic foot usually taken at 115 pounds.

b = reduced section of brick work $= B\left(1 - \frac{B}{D}\right)$

S = diametrical section of chimney = mean diameter $\times$ total height in feet; and,

P = pressure in pounds per square foot, to overturn chimney.

$$P = \frac{(.33 - q')\ W\ D}{H\ S} \text{ for square chimneys.}$$

$$P = \frac{(.25 - q')\ W\ D}{H \frac{S}{2}} \text{ for round chimneys.}$$

$$W = 4\ w\ b\ S \text{ for square chimneys.}$$

$$W = 3.1416\ w\ b\ S \text{ for round chimneys.}$$

Suppose a chimney, the shaft of which has a diameter of 10 feet at base and 8 feet at top, and a height $= 2\ H$ of 75 feet, and the total pressure at base is 150,000 pounds, what pressure of wind per square foot of surface will be necessary to overturn it; or, rather, what will be the stability in pounds pressure per square foot of effective surface.

H = (roughly) 37.5 feet.

W = 150,000 pounds.

$q' = 0.$

D = 10 feet.

$$S = 75 \times \left(\frac{10 + 8}{2}\right) = 675 \text{ square feet.}$$

Then—

$$P = \frac{.33 \times 150{,}000 \times 10}{37.5 \times 675} = 19.555 \text{ pounds for square chimney, and}$$

$$P = \frac{.25 \times 150{,}000 \times 10}{37.5 \times \frac{675}{2}} = 29.63 \text{ pounds for round or octagon chimney.}$$

Omitting the single item of cost, round or octagonal chimneys are to be preferred to square ones as offering a greater stability and draught efficiency for a given cross section and height, and as presenting a more sightly appearance.

Mr. Bourne offers the following formulæ for cross section of chimney (flue or core):

$$\frac{C\,12}{\sqrt{h}} = A = \text{cross section of flue in inches.}$$

Where C = coal in pounds burned in entire grate per hour, and h = height of chimney from surface of grate.

A chimney 90 feet high connected with a boiler having 60 square feet of grate surface burning 15 pounds of coal per square foot of grate per hour, according to Bourne, should have a minimum cross section of flue, of

$$\frac{900 \times 12}{\sqrt{90}} = 1139.2 \text{ square inches.}$$

The author thinks above dimensions too small for good results, and suggests the following formula as representing his practice for bituminous coal, at average rates of consumption for natural draught (15 to 25 pounds per square foot of grate per hour):

$$A = \frac{1.8\,g}{\sqrt{h}}$$

Where A = area of chimney flue, in square feet, at smallest section, g = area of grate surface in square feet, and h = effective height of chimney in feet. Applying this formula to above data, the area of flue becomes—

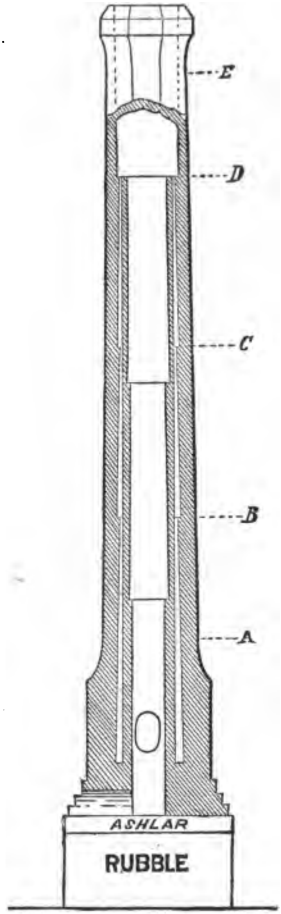
$$A = \frac{1.85\,g}{\sqrt{h}} = \frac{1.85 \times 60}{\sqrt{90}} = 11.71 \times 144 = 1686.24 \text{ square inches.}$$

The forms in cross section generally adopted are square, round, or octagon.

Sheet iron chimneys are to be avoided, excepting for temporary uses. Iron chimneys, however, with an outer and inner shell, and a non-circulating jacket space between, will give better results in efficiency than brick chimneys of same height and diameter; but will not compare with the latter for strength and durability.

The following table from Smeaton, gives the pressure in pounds per square foot of perpendicular surface for different gales of wind:

Velocity, miles per hour.	Pressure.	Velocity, miles per hour.	Pressure.
1	.005	20	2.000
2	.020	25	3.125
3	.045	30	4.500
4	.080	40	8.000
5	.125	50	12.500
10	.500	60	18.000
12½	.781	80	32.000
15	1.125	100	50.000



The figure is a reduced vertical section of an octagon chimney. designed by the author for the Cincinnati Gas Light and Coke Company, for two sectional boilers of 900 square feet of heating surface each; burning coke breeze.

The following are the principal dimensions:

Height from boiler room floor to top	91 ft. 6″
Depth of foundation	10 ft. 0″
Least cross section of flue	12 sq. ft.
Thickness of shaft *A* to *B*	21 ″
" " " *B* " *C*	16.5 ″
" " " *C* " *D*	12.75″
" " " *D* " *E*	9. ″

QUANTITIES OF MATERIAL.

Brickwork, bricks	105240
Ashlar courses, foundation, perches	19.44
Rubble work, foundation, perches	110.16

ESTIMATED BASE LOADS.

Per sq. ft. of brickwork	1.634 tons.
" " " foundation	1.590 "
" " " core	2.635 "

FURNACES AND BOILERS.

PERFORMANCE.

Experimental data on the conditions calculated for maximum economy in the performance of boilers and furnaces are very limited, and what we have, by no means reconcile the various opinions that have for years prevailed upon the subject of boiler and furnace construction. From Mr. Pole we have the statement that the average performance of Cornish boilers thirty years ago, was 10.75 pounds per pound of coal, with Welsh coals. We have many varieties of coal in the United States that are equal to the Welsh coal, and the average evaporation of American boilers is considerably less than *eight pounds* per pound of coal. The care with which a boiler is set and operated has much to do with the consumption of fuel, and perhaps the low cost of coal in many localities has made boiler constructors indifferent to the economy of performance. However this may be, there can be no good reason why the development of boiler and furnace economy should not keep pace with the improvement of the engine.

According to Mr. D. K. Clark, in discussing boiler and furnace economy, "the efficiency decreases directly as the grate surface—increases as the square of the heating surface (with the same area of grate and efficiency of fuel); the necessary heating surface increases as the square root of the performance, or for a fourfold performance a twofold heating surface is required. The heating surface also increases as the square root of the grate with the same efficiency of fuel; thus, if the grate area be increased four times, the heating surface should be doubled."

From numerous experiments on locomotive boilers it appears that the ratio of heating to grate surface can never be in excess, while it may be too low for average economy. In fire-box boilers, when the hot gas passes through a set of horizontal tubes to the chimney, such as a locomotive or portable boiler, nearly 60 per cent of the evaporation is due to the heating surface surrounding the fire-box, and only 40 per cent to the tubes. In the ordinary portable boiler for farm use the heating surfaces surrounding the fire-box furnishes over 75 per cent of the evaporation.

From Peclet's deductions, it appears that the course of the hot gas should be from above downwards. Dr. Pole entertains the same opinion. The Cornish and Lancashire boilers carry out this principle, the coal being charged into furnaces placed at the forward end of the flues or tube, the hot gas passing aft through the tube,

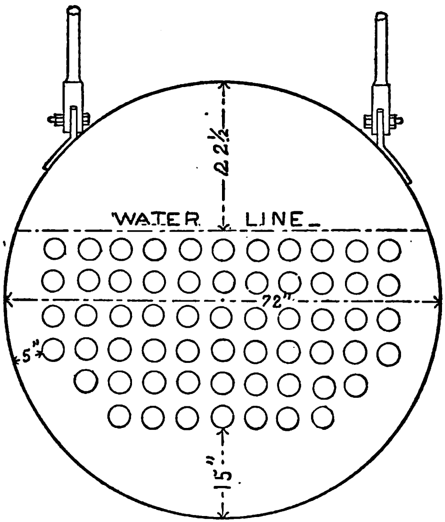
thence down and forward under the shell of the boiler, thence to the chimney. Fire-box drop flue boilers are similarly constructed, except the hot gas passes aft through an upper series of tubes, passes forward through a lower series of tubes, and then passes back under the shell, making nearly three lengths of the boiler in its circuit.

With the ordinary return flue boiler, such as are largely in use in the West, length seems to regulate the economy of performance. Referring to the table of boiler and furnace performance, the J. W. G. & Co. boilers were set in miserable furnaces; the bridge walls were broken down, and the side walls cracked and leaking; and by test, at least 12 per cent more of the colorific value of the fuel could have been utilized, by reducing the temperature of waste gas (as it passed into the chimney) to 500 degrees Fahr. The lack of bridge wall, and failure to provide against radiation in the side walls of furnace, entailed a farther loss of 10 per cent; whence the evaporation in this case would become $\frac{8.365}{.78} = 10.72$ pounds, per pound of coal. The E. D. A. & Co. furnace and boiler were in excellent condition, and the evaporation of 8.307 pounds is a maximum for an equivalent arrangement.

It is not possible to furnish laws that will apply to the performance of boilers already in use, or to be used for the construction of furnaces and boilers in the future; as experience has shown that too many elements beyond qualification are embodied in the problem. But the following general suggestions may be useful to those having occasion to construct new boilers.

Horizontal tubular boilers are to be preferred for economy, but, when used with bituminous coal, the tubes must be attended to frequently, to avoid accumulation of soot, the detrimental effect of which is *dual;* first, in diminishing the effective heating surface, and second, in diminishing the effective draught, by the largely increased frictional resistance of the sooted surfaces. The predjudice entertained by some steam users (upon the score of safety) against tubular boilers is purely chimerical, a properly designed tubular boiler of same dimensions and material of shell, being in all respects as safe as a flue or cylinder boiler. In banking the tubes in a tubular boiler, care should be had to give ample space between tubes, and between the tubes and shell, for cleaning. The tubes should nowhere approach the shell closer than 5 or 6 inches, and a clear space of 14 to 16 inches should be allowed under the lower row of tubes.

The figure is a reduced transverse section, of horizontal tubular boiler of Otis steel, designed by the author for the Carlisle Building of Cincinnati.



LENGTH, 18 FEET.

Vertical tubular boilers are very wasteful of fuel, and should never be adopted to furnish steam for engines of any magnitude. When a boiler is very limited in length, then the fire-box drop flue style will be found to give the best results. This pattern, however, should never be used with bituminous coal, unless the combustion be practically perfect, as the rapid deposit of soot would destroy the efficiency, and render it very wasteful of fuel.

The capacity of a boiler should be expressed in its evaporation per square foot per hour. The term H. P. has no application to a steam boiler, from the fact that what would be a twenty H. P. boiler with one engine, might be sixty H. P. with another engine. The evaporation per square foot of heating surface varies in different forms of boilers. The maximum obtained by the author with return flue boilers is 6 pounds. The average, however, is about 3 pounds.

A boiler 20′ long, 42″ diameter, 2—15″ flues, has about 300 square feet of heating surface; and, with an evaporation of 3 pounds per square foot, would furnish 900 pounds of steam per hour. With a first-class slide valve engine, well proportioned to its load, the water (steam)

per H. P. per hour would be 45, hence capacity of boiler $\frac{900}{45} = 20$ H. P. If the boiler was connected with a Harris-Corliss Engine using 25 pounds of steam per H. P. per hour, then its capacity would be increased to $\frac{900}{25} = 36$ H. P. This boiler should have from 10 to 12 square feet of grate surface, burning about 10 pounds of coal per square foot of grate per hour. Suppose we require the dimensions of heating and grate surface for a pair of boilers and furnace, to furnish steam to an engine of 100 H. P., using 45 pounds of steam per H. P. per hour. The average of American coals will, with well porportioned boilers and furnaces, furnish 9 pounds of steam per pound of coal; hence coal burned per hour $\frac{100 \times 45}{9} = 500$ pounds; and, with a consumption of 15 pounds per square foot of grate per hour, we should have 33.33 square feet, or a grate 4.5′ deep × 7.5′ wide. Assuming the heating surface capable of evaporating 3 pounds per square foot per hour, the combined heating surface of two boilers should be 1,500 square feet, and the ratio of heating to grate surface becomes 45.

Furnaces and boilers should always be adapted to the location, fuel to be burned, and economy of engine; and it will always be profitable to those desiring new boilers and furnaces to have plans for both from a competent engineer.

The performance of a steam boiler is usually estimated on the conversion of water into saturated steam, from 212° Fahr. and at pressure of atmosphere. Thus we reconcile the differences in temperature of feed water and evaporation. The coal burned is always an uncertain element, and a proper test of a boiler is to base the efficiency on the combustible. When the test is of the efficiency of the coal, then the evaporation should be based on the total coal burned to gas, or ash, and no allowance should be made for non-combustible.

Steam boilers, except when the waste heat from blast or puddling furnaces is utilized for making steam, are always worked in conjunction with a furnace of some description, and it is customary to consider the performance of boiler and furnace as a whole. The function of the furnace is to produce the largest percentage of carbonic acid from a given weight of fuel, and with the greatest possible elevation of temperature of the products of combustion. Every pound of air in excess of that necessary for combustion that enters the furnace and passes out of the chimney, takes up a certain quantity of heat that otherwise would be utilized in making steam, and diminishes the temperature.

The function of the boiler is to absorb and transmit to the water the

heat due to the action of the furnace. When a test is simply one of relative efficiency of boiler; then, when it is practicable, the same furnace should be used; and when it is a test of relative efficiency of furnace, the same boiler should be used. In marine and fire-box boilers the design is continuous, and these distinctions can not successfully apply. But when tests are for ultimate efficiency, then they should be conducted in such a manner that the performance of the boiler may be separated from that of the furnace; and conversely we should be able to estimate the performance of furnace, independent of the performance of boiler.

It has been the custom, up to a very recent period, to estimate the efficiency of boilers upon the quantity of water pumped in. But all such tests, with our present knowledge, are worthless, as the primage is one of the most important factors in the problem. Every boiler should be designed to furnish saturated steam; and when the boiler is incompetent to do this, then a steam-chimney should be added, and the dryness limited to saturation, or a few degrees above.

Furnaces using previously heated air for combustion are to be preferred, when no loss of heat is occasioned in elevating the temperature of the air.

Smoke-prevention, in furnaces burning bituminous coal, has long been a favorite scheme with inventors; but it is extremely doubtful if success in this direction will ever be attained. Smoke-prevention, while within the bounds of possibility, is beset by so many obstacles that the task of attempting it is almost as much of an *ignus fatuus* as the *mobile perpetuum*.

The supposition that smoke is an evidence of imperfect combustion is only partially true, as many English experiments on furnaces show that the loss of efficiency is very small with an intelligent working of the fires, and, in many cases, almost inappreciable. Chemical analysis of the products of combustion, of well designed steam boiler furnaces, properly worked, has shown that the percentage of carbonic oxide is small, and the proportion of free carbon too minute to be of any practical value.

It is not difficult to construct a furnace that will give good results with anthracite coal, as we have but a single combustible element to deal with. But with bituminous coal we have the volatile matter, and the carbon to work; and a furnace properly adapted to work the gases can not be equally efficient with the carbon, and conversely a furnace calculated for maximum efficiency of carbon will yield but indifferent results in the combustion of the gases.

Furnaces for bituminous coal, upon the oven principle, when combustion is effected under a fire brick arch and out of contact with the boiler, are moderately successful in the prevention of smoke, but are objectionable, owing to the exalted temperature of the hot gas impinging upon the shell or tubes of the boiler.

FURNACE AND BOILER TRIALS.

FLUE BOILERS.

Location.	Date.	Designation	Boilers.	Heat'g surface.
Cincinnati..	1875	Ashcroft	1—40″ × 22′—2—14″ flues.........	337.74
" ..	"	Baum........	1—40″ × 22′—2—14″ flues.........	288.42
New York...	"	Hoyt	Cylinder drop flue...............	530.00
" ...	"	"	" " "	530 00
" ...	1876	"	" " "	530.00
Cincinnati..	1877	J.W.G. & Co.	4—46″ × 32′—2—17″ flues, each. / 3—46″ × 32′—2—16″ flues, each.	2166.72 / 1274.79
" ..	"	E.D. A. & Co.	2—48″ × 20′ 2—10″ flues / 4— 8″ flues each..	882 80
" ..	"	McN. & U...	1—48″ × 28′ 2—10″ flues / 4— 8″ flues	624.58
Indianapolis	"	G. & Co......	2—54″ × 20′ 2—11″ flues / 4— 8″ flues each..	931.08
"	"	"	2—54″ × 20′ 2—11″ flues / 4— 8″ flues each..	931.08
Hamilton....	"	Ordinary ..	1—48″ × 30′—2—18″ flues.........	520.70
"	"	Jenks........	1—48″ × 30′—2—18″ flues.........	520.70
Cincinnati..	"	M. F. & Co..	2—48″ × 24′ 1—14″ flue / 2—10″ flues / 2— 9″ flues each..	1056.31
" ...	"	Moerlein....	2—42″ × 24′—2—14″ flues, each..	683.70
" ..	"	Fisher.	1—48″ × 24′ 2—10″ flues / 4— 8″ flues	519.45
Bethalto	1879	M. & G......	3—42″ × 26′—4—10″ flues, each...	1355 77
" ...	"	"	3—42″ × 26′—4—10″ flues, each...	1355.77
Alton, Ills...	"	D. R.S. & Co.	2—48″ × 26′ 2—14″ flues / 2—15″ flues each..	1201.47
" ...	"	"	2—48″ × 26′ 2—14″ flues / 2—15″ flues each..	1201.47
Waterloo, Ill	"	C. & E.......	5—39″ × 24′—2—14″ flues, each..	1480 32
St. Louis...	"	A. M. Co.....	5—48″ × 26′—4—14″ flues, each..	2764.16
Cincinnati..	"	C. W. W....	2—48″ × 24′ 2—10″ flues / 4— 8″ flues each..	1082.98
" ...	1880	Warden......	2—48″ × 24′ 2—10″ flues / 4— 8″ flues each..	1082.98
" ...	"	Hutchinson.	2—48″ × 24′ 2—10″ flues / 4— 8″ flues each..	1082.98
Evansville..	1881	E. W. W.....	2—48″ × 16′—12—6″ flues, each	932 01
" ..	"	"	2—48″ × 16′—12—6″ flues, each..	932 01
Newport	1882	S. I. & S. W..	2—48″ × 28′—2—16″ flues, each..	895.84
"	"	Gearing.....	2—48″ × 28′—2—16″ flues, each..	895.84

FURNACE AND BOILER TRIALS.

FLUE BOILERS.

Grate surface.	Ratio heating to Grate surface	Coal per sq. ft. of Grate per hour.	Steam per sq. ft. of heat'g surface per hr.	Coal Burned.	Steam per p'nd of coal from and at 212	Authority.
12.41	27.210	24.174	5.046	Straitsville, Ohio.	7.761	Expert's report.
20.16	14.300	12.400	4.534	"	7.062	"
15.00	35.333	13.93	4.50	Maryland Coal.	10.640	Skeel.
15.00	35.333	19.10	4.92	American Cannel.	9.36	"
15.00	35.333	16.8	4.72	Maryland Coal.	11.200	"
82.27	26.337	16.516	4.844	Pittsburgh, No. 2.	8.365	Author.
60.62	21.292					
38.00	23.23	....	....	"	8.307	"
20.25	30.840	....	....	"	7.704	"
38.00	24.520	11.34	2.115	Highland, Ind.	5.212	"
38.00	24.520	15.84	3.159	"	5.240	"
22.50	23.360	....	....	Pittsburgh, No. 2.	4.770	"
22.50	23.360	....	....	"	6.831	"
24.00	44.013	....	....	"	7.258	"
37.58	18.195	....	....	"	7.875	"
16.64	31.97	11.214	1.926	"	4.828	"
51.00	26.58	26.66	5.62	Bethalto Ill.	6.000	"
51.00	26.58	16.47	3.40	"	6.030	"
41.83	28.72	25.590	4.364	Illinois.	5.380	"
41.83	28.72	17.21	3.581	"	6.446	"
79.32	18.662	14.072	5.670	Belleville, Ill.	8.104	"
85.31	32.405	24.114	4.583	"	6.725	"
19.04	56.86	21.010	3.718	Pittsburgh, No. 2.	10.032	"
16.90	64.08	36.982	4.971	"	9.705	"
16.90	64.08	36.257	5.403	"	10.149	"
45.00	20.711	8.001	2.858	Anthracite.	8.432	"
45.00	20.711	8.255	2.903	"	8.251	"
35.00	25.60	25.959	6.390	Pittsburgh, No. 2.	6.252	"
35.00	25.60	15.326	4.695	"	8.465	"

FURNACE AND BOILER TRIALS.

TUBULAR BOILERS.

Location.	Date....	Designation.	Boilers.	Heat'g surface.
New York...	1871	Am. Instit'te	Lowe.	913 00
" ...	"	"	Blanchard..................	440 00
Cincinnati...	1879	Walker.. ..	1—60″ × 16′—48—4″ tubes.......	963.63
" ..	"	Eureka......	2—38″ × 16′—21—4″ tubes........	880.15
" ..	"	Price........	1—54.5″ × 16′—40—4″ tubes.....	823.05
" ..	"	Murphy.....	1—36″ × 8′—30—3″ tubes........	327.79
Milwaukee..	"	M. M. Co....	2—54″ × 16′—39—4″ tubes.. ..	1605.13
" ..	"	Germain ..	1—38″ × 10′—39—2.5″ tubes......	325.95
La Crosse...	"	A A. F. & Co.	2—60″ × 12′—50—4″ tubes.. ...	1536.92
Natchez.....	1881	N. C. M	3—56″ × 16′—47—4″ tubes..	2823.55
Chicago.....	1882	N K F. & Co	1—54″ × 10′—44—3 5″ tubes......	758.17
Saratoga ...	"	S.W.W. ...	2—66″ × 18′—87—3″ tubes.... ...	2957.50

LOCOMOTIVE BOILERS.

Cincinnati...	1874	Exposition..	2—Locomotive—East—boilers.	982.84
" ..	"	" ..	" West "	982 84
" ..	1878	C.H.&D.R.R.	Baldwin Standard 4 driver.	898.67
Hamilton....	"	"	"	898.67
Twin Creek..	"	"	"	898.67
Cincinnati...	1879	Sulter.......	Locomotive.	100.00
" ..	1880	C T D & Co.	"	288.75
" ..	"	Fisher.......	"	300.70
" ..	"	"	"	300.70
Vincennes ..	"	O. & M. R. R.	Rogers Standard 4 driver.	984.33
Ludlow......	1882	C. S. R. R...	Baldwin Standard 4 driver.	1073.01
"	"	" ...	"	1073.01
Cincinnati...	1882	C.G.L.&C.Co.	3 Locomotive boilers.	1768.95
" ...	"	"	"	1768.95

TUBULOUS SAFETY BOILERS.

New York...	1871	Am. Instit'e.	Root............................	876.50
" ...	"	"	Allen..........................	920 00
" ...	"	"	Phleger..	600.00
Cleveland...	1877	A. & Co....	Root.	2805.96
Chicago.....	1882	N.K.F. & Co.	Firmenich	1632.97
"	"	"	"	1632.97
Cincinnati..	"	M. O. W......	2 Babcock & Wilcox.............	1679.00
" ..	"	M. D. Co.....	"	2640.00

FURNACE AND BOILER TRIALS.

TUBULAR BOILERS.

Grate surface..	Ratio heating to Grate surface..........	Coal per sq. ft. of Grate per hour..........	Steam per sq ft. of heat'g surface per hr...	Coal Burned.	Steam per p'nd of coal from and at 212 ..	Authority.
37.75	24 200	9 71	3 10	Buck Mountain.	10 400	Thurston.
8 50	51 800	12 10	1 92	"	11 340	"
28 33	34 015	13.413	2.760	Pittsburgh, No. 2.	7 000	Author.
24.00	36 673	14 250	3 131	"	8.392	"
22 50	36.58	11 973	3 894	"	11.898	"
10.50	31.22	7.419	2 958	"	12 450	"
46 12	34 80	10 95	2 391	Massillon.	8 905	"
10 00	32 59	3 243	1 064	Briar Hill.	11 416	"
51.75	29 70	10.300	3 022	Wilmington, Ill.	9 639	"
62 92	44.87	10 160	1 773	Pittsburgh, No. 2.	8.358	"
18.00	42 12	22 806	3.421	Erie Coal.	6 789	"
57.00	51 89	5 883	1.193	Lackawanna.	11.286	"

LOCOMOTIVE BOILERS.

25.40	38 700	12 303	1 908	Pittsburgh, No. 2.	6 665	Author.
25.40	38 700	11 071	2 377	"	7.167	"
15 09	59.647	83 913	9 963	"	8 360	"
15.09	59 647	171.822	13 015	"	5 344	"
15 09	59 647	117.272	12 241	"	7 300	"
1.983	50.43	33 31	5.52	"	9 250	"
11.332	25 903	43 053	8.493	"	5 024	"
7.216	41 670	41.343	9 386	"	8 820	"
7 216	41 670	50 945	9 774	"	8 001	"
13 91	70 764	146 288	9.515	Washington, Ind.	4 605	"
15 05	71.297	66.131	6.417	Hocking Valley.	7.957	"
15 05	71 297	72.773	6.911	"	7 905	"
55 63	31.799	18.601	3 31	Coke—Breeze.	6 005	"
55.63	31.799	17.836	3.544	Kanawha Slack.	6.486	"

TUBULOUS SAFETY BOILERS.

27.00	32 500	11.73	2 65	Buck Mountain.	10 640	Thurston.
32 25	28 500	13 88	3 59	"	10.600	"
·23.00	26 100	10.13	2.83	"	10 490	"
96.77	29 000	....		Washingtonville.	5 795	Author.
80 50	53 54	19 562	2.384	Erie Coal.	7 037	"
80 50	53 54	16 665	1 940	"	6 633	"
49.83	33 692	18.477	3 689	Pittsburgh Coal.	7 877	"
43.74	60.352	27.433	4.042	"	9.570	"

DUTY OF PUMPING ENGINES.

The term "duty" is a measure of the efficiency of an engine, and is based upon the delivery of water into the head (plus the friction of the rising pipes) per hundred pounds of coal. It is customary to express duty in foot pounds.

The method usually employed neglects the actual delivery of water, and head, against which the pump works, but assumes that the area of the pump piston, × the average pressure or head pumped against measured to level of water in the pumping well (and the pressure due friction), × the lineal travel of the piston, represents the work done, and this divided by one pound of coal for each hundred burned represents the duty; or, by formula,

$$D = \frac{A \times P \times F}{C} \times 100$$

when A = area of pump piston, P = load in pounds pressure per square inch, F = stroke of piston in feet into twice the revolutions or double strokes, C = coal consumed for travel of piston (F).

The following data is from contract trial of Simpson compound pumping engine, built by E. P. Allis & Co., for the city of Milwaukee. Diameter of pump, 3.33 feet; stroke, 7 feet; revolutions, 39,143; load per square inch of piston, 72.503 pounds; and coal fired, 64,750 pounds. The duty, by calculation, becomes

$$\frac{1254.13 \times 72.503 \times 548{,}002}{647.50} = 76{,}955{,}720 \text{ foot pounds.}$$

This method is employed in estimating the duty when the engine pumps directly into the mains, or into a stand-pipe. When the delivery of water is into a reservoir, the following method is employed.

The delivery of water into the reservoir is noted either by weir measurements, or by calculating cubic contents of reservoir at beginning and at end of trial, or by estimating theoretical delivery of pumps, and allowing a uniform slip (to be determined by experiment).

When the delivery of water is very regular, or subject to slight fluctuations, the weir measure is the most delicate test of discharge, and when several engines are delivering into the same reservoir at the same time, the weir measurement is absolutely necessary. When the discharge is determined by measurements of the reservoir at beginning and at end of trial, previous and subsequent observations should be made of the loss of water by leakage and surface evaporation, and the discharge from force main corrected accordingly.

When the actual delivery of water is made the basis for estimating the duty, the lift is taken, either by difference of levels of water in pump well and reservoir, or by taking the pressure on the rising main in the engine house, and adding the difference of levels between the gauge and water in the well; to this is added an allowance for frictional resistances between the gauge and well. The delivery is usually reduced to gallons, and the weight of water at mean observed temperature, accurately determined. Then, by formula,

$$D = \frac{G \times W \times H}{C} \times 100$$

where G = discharge in gallons during trial, W = weight per gallon, H = constant head in feet to which the water is delivered, and C = coal burned, as before.

The following data is from the contract trial of the Lawrence, Mass., Pumping Engine (Leavitt, compound), built by I. P. Morris & Co., Philadelphia:

Discharge by weir measurement, 4,527,340 gallons.

Weight per gallon, 8.38 pounds.

Lift, including allowance for friction, 175.47 feet.

Coal consumed, 7,266 pounds.

$$\frac{4,527,340 \times 8.38 \times 175.47}{7,266} \times 100 = 91,620,912$$

to which add 5 per cent (contract allowance for slip), when the duty becomes,

96,201,956.84 foot pounds.

Another method of estimating the duty is to determine the mean resistance against which the pump works (including vacuum necessary to lift the water from the pump well), by indicator diagram. This constitutes the lift. The delivery of water may be determined by actual measurement, when this is practicable, or by calculating the capacity of the pump, and deducting assumed slip. The slip, or loss of action of the pump (being the difference between the calculated and actual delivery).

The contract allowance for frictional resistances of water passages into and out of pumps ranges from one to two pounds.

An allowance of one pound (2.308 feet), for frictional resistances of water passages into and out of pump, is ample for well constructed waterways; but there are many instances where the volume of flow and water passages are so badly proportioned that a resistance of several pounds is occasioned by the friction of water entry and exit.—(See remarks on Warden compound engine.)

LOSS OF ACTION OR SLIP OF PUMPS.

(In percentage of calculated delivery.)

LOCATION.	ENGINES.	SLIP.	AUTHORITY.
Cincinnati.	Combination Engine.	8.73	Hermany.
"	Harkness "	6.60	"
"	Powell "	8.54	"
"	Redemption "	7.96	"
"	Warden Compound Engine.	7.693	Hill.
"	" "	7.591	Hill.
Trenton.	Wright "	3.58	Slade.
Lynn.	Leavitt. "	3.99	Worthen.
Milwaukee.	Hamilton "	2.26	"
Memphis.	Gaskill " No. 1.	2.43	Hill.
"	" " No. 2.	2.44	"
Providence.	Corliss " Pettaconsett.	0.50	Gray.
Troy.	Holly & Gaskill Engine.	3.80	Greene.
Buffalo.	Worthington Comp. Engine.	7.34	Hill.
Philadelphia.	" "	1.50	Board of Experts.
Lowell.	" "	2.25	Evans.
"	Simpson Compound Engine.	2.52	Board of Experts.
Lawrence.	Leavitt "	5.23	Worthen.
Brooklyn.	Engine No. 1.	2.00	Kirkwood.
"	" No. 2.	1.50	"
"	" No. 3.	2.50	"
Salem, Mass.	Worthington Comp. Engine.	3.125	Journal Am. Soc. Civil Engineers.
Providence	" "	2.50	"
Jersey City.	Cornish Beam Engine.	9.14	"
Hartford.	Single Cyl. "	6.20	"

EFFICIENCY OF PUMPING ENGINES.

The following data is from the experiments of M. Tresca, Paris, upon a double-acting piston pump, containing two barrels connected at the bottom by a water passage, and each piston provided with a single series of valves. Those of the first piston opening downward, and those of the second piston opening upward; the water entered at the top of the first cylinder, passed downward through the first piston, thence upward through the second cylinder and piston, and out at the top of the second cylinder. The pistons were each 18 inches diameter and of 6 inches stroke.

The pump was worked at different rates of speed, and under pressures (heads) ranging from 1 to 5 4. The efficiency, and ratio of water discharged to calculated displacement of pistons, being observed for the several speeds and heads.

It will be observed that Tresca has proven by experiment what was previously believed to be true—that the efficiency of pumping engines is directly as a function of the head, and that the loss of action was no greater at moderately high speeds than at low speed, and was practically unaffected by the head pumped against.

It is evident that as the frictional resistance of a steam engine (omitting extra friction due to load) is a constant quantity for any given speed without regard to load—that the percentage of this loss is a constantly diminishing quantity (within reasonable limits) with increase of load. And that pumping engines, with ample strength and wearing surfaces, should increase in efficiency (duty), with increase of head.

Revolutions per minute.	Total Head Feet.	Efficiency.	Ratio of Discharge to Displacement of Pump.
33.00	14.10	43.1	96.0
42.40	14.10	43.1	97.2
55.08	14.10	44.7	92.0
60.55	16.63	53.7	94.5
Averages......	14.73	46.1	94.92
23.75	23.22	63.7	....
45.48	24.93	53.0	95.7
60.00	27.32	53.0	95.4
Averages......	25.16	56.6	95.55
39.62	33.54	66.7	97.6
43.75	33.54	69.0	98.1
40.50	33.39	61.2	91.4
55.00	35.55	63.2	95.4
28.00	35.55	71.4	91.2
Averages......	34.31	66.2	94.74
31.00	42.80	73.6	93.9
24.33	45.62	73.7	89.8
52.68	45.62	71.0	95.3
32.50	46.28	66.5	91.7
55.00	46.97	70.4	94.8
50.00	49.33	71.0	95.8
61.98	51.00	68.7	90.5
55.00	75.44	70.4	92.5
Averages......	50.38	70.7	93.04

FRICTIONAL RESISTANCE OF WATER PASSAGES INTO AND OUT OF PUMPS.

This load or head which ranges in contracts for pumping engines for public water supply—from *one* to *two* pounds—is the difference between the apparent head pumped against as measured from the source of supply, and the net absolute head as read from an indicator diagram.

The head in the suction pipe may be taken either by a vacuum gauge,

or pressure gauge, dependent upon circumstances; if the water is lifted from a well—by a vacuum gauge, and if taken under pressure as from an elevated reservoir—by a pressure gauge.

Suppose the barometer reads 30 inches, or 14.727 pounds, and the vacuum gauge on suction pipe of pump indicates 15 inches, or

$$14.727 - 7.3635 = 7.3635 \text{ pounds,}$$

absolute head; and pressure gauge on discharge main near pump indicates 75 pounds, then apparent head pumped against, is

$$(75 + 14.727) - 7.3635 = 82.3635 \text{ pounds.}$$

Suppose the absolute head (as read from the indicator diagram), upon suction side of pump, is 14 inches or 6.8723 pounds, and absolute head upon discharge side of pump 90 pounds, then total head pumped against is

$$90 - 6.8723 = 83.1277 \text{ pounds,}$$

and frictional resistance of water passages is

$$83.1277 - 82.3635 = .7642 \text{ pound, or } 1.7638 \text{ feet,}$$

of which loss

$$7.3635 - 6.8723 = .4912 \text{ pound}$$

is friction of entry, and

$$90 - (75 + 14.727) = .273 \text{ pound}$$

is friction of exit.

(From Author's Report on Warden Compound Engine.)

"From a series of twenty-five diagrams from the upper end, and twenty-five diagrams from the lower end of pump driven by the high pressure engine, taken during the last four hours of the trial, it appears that the mean pressure upon the pump piston was 123.32 pounds per superficial inch of exposed surface, corresponding to a water head of

$$123.32 \times 2.308 = 284.62 \text{ feet.}$$

"During the interval when water diagrams were taken, the pressure gauges on the suction and force pipes were read every minute, from which is deduced as a mean head on force pipe

$$(136.5 \times 2.308) + 12.5 = 327.54 \text{ feet,}$$

and on the suction pipe

$$(22.5 \times 2.308) + 12.5 = 64.43 \text{ feet,}$$

and net head pumped against during the time high pressure (engine) water diagrams were taken, as measured in the force main to the center of pump cylinder, was

$$327.54 - 64.43 = 263.11 \text{ feet,}$$

and pressure per superficial inch of pump piston required to open

the suction and delivery valves, and overcome the frictional resistance of water passages into and out of the pump, becomes

$$\frac{284.62 - 263.11}{2.308} = 9.32 \text{ pounds.}$$

Of this pressure

$$27.916 - 21.56 = 6.356 \text{ pounds,}$$

was expended in lifting the suction valve and overcoming the friction of entry, and

$$144.88 - 141.916 = 2.964 \text{ pounds,}$$

was expended in opening the delivery valve and overcoming the friction of exit.

"Twenty-five diagrams also were taken from each end of the pump worked by the low pressure engine, during the last four hours of the trial, from which is obtained, as the mean pressure per superficial inch of pump piston,

$$128.45 \text{ pounds,}$$

corresponding to a water head of

$$128.45 \times 2.308 = 296.46 \text{ feet.}$$

"The mean readings of pressure gauges on water mains during the interval of time, whilst low pressure (engine) water diagrams were taken, were for suction pipe 22 pounds, and for force pipe 137 pounds, from which is deduced, as a mean head on the force pipe

$$(137 \times 2.308) + 12.5 = 328.69 \text{ feet,}$$

and on the suction pipe

$$(22 \times 2.308) + 12.5 = 63.27 \text{ feet;}$$

and net head against which water was pumped during the time water diagrams from low pressure (engine) pump were taken, as measured in the force main to center of pump cylinder, becomes

$$328.69 - 63.27 = 265.42 \text{ feet,}$$

and pressure per superficial inch of pump piston required to open the suction and delivery valves, and overcome the frictional resistance of water passages into and out of the pump, was

$$\frac{296.46 - 265.42}{2.308} = 13.45 \text{ pounds,}$$

of this pressure, $27.416 - 18.60 = 8.816$ pounds

was expended in lifting the inlet valve and overcoming the friction of entry, and

$$147.05 - 142.416 = 4.634 \text{ pounds}$$

was expended in lifting the outlet valve and overcoming the friction

of exit. The usual allowance is *one* pound pressure per superficial inch of pump piston for overcoming frictional resistances in the pump, and in moving the valves; or about $\frac{9}{100}$ of the pressure required in the pumps of this engine.

"The relative thickness of rubber valves in use in these pumps, made necessary by the head against which the pumps work, together with the cramped arrangement of inlet and outlet connections are responsible for the serious loss of power in filling and discharging the pumps.

CAPACITY TESTS OF PUMPING ENGINE.

The following matter is quoted from the author's report to the Water Company of Memphis, Tennessee, and the water commissioners of Buffalo, New York, upon the capacity performance of the Gaskill and the Worthington compound pumping engines, respectively:

Gaskill Compound Pumping Engine.

The contract provides that each engine shall be capable of pumping 4,000,000 gallons in twenty-four (24) hours, at a piston speed of one hundred and fifty-five (155) feet, and that this work shall be done easily, without overstrain of any part of the machine.

The specification provides that this quantity of water shall be delivered against a head as indicated upon the water pressure gauge of sixty-five (65) pounds, and that the discharge shall be measured over a weir.

The original specification provides that the vertical distance from the engine room floor to low water mark shall be forty-two (42) feet, and vertical distance from said datum to center of water pressure gauge shall be six (6) feet, or total difference of low water mark and water pressure gauge forty eight (48) feet.

In the construction of the pump house the engine room floor was elevated 72.36 feet above low water mark, and the water pressure gauge was located 8.28 feet above engine room floor, making total distance from center of water pressure gauge to low water mark 80.64 feet, or 32.64 feet higher than provided in the original specification The difference in elevation equivalent to a pressure of 14.2 pounds per square inch must be deducted from the pressure by gauge against which the engines are required to pump by the specification, in order that the actual head pumped against for capacity test shall equal the head provided by the terms of contract.

The minimum gauge pressure for capacity tests was accordingly

fixed at fifty-one (51) pounds, which pressure was obtained by partially closing a stop valve in the discharge pipe.

The engines pump into the mains upon the direct supply system, and the cutting of the principal distribution main for the purpose of weir measurements involved a stoppage of the machinery for several days and a corresponding loss of water to the consumers; upon consultation with the water company and the contractor, it was decided to abandon the weir measurements, and test the capacity of the engine by pumping into the small reservoir at the pumping station; in furtherance of this plan the distribution main was cut, and a new stop valve inserted beyond the branch leading to the reservoir, in order that all leakage should be confined to the reservoir proper and its immediate connections.

The reservoir was measured for the purpose of the capacity trials, and found to have the following dimensions at the surface of the banks:

Length, mean of both sides	255 6	feet.
Width, mean of both ends	130.925	"
Depth	15 775	"
Angle of inside slope	35° 45′	

The corners of the reservoir are 90° arcs of circles to which the sides and ends are tangent, with a radius of 19 feet at the surface of the banks, and 0 at the bottom of the slope, where the horizontal section is a true rectangle.

To determine the leakage of the reservoir, all connections therewith were closed, and the level carefully taken at 3:00 P. M., January 8th, and again at 5:00 P. M., two (2) hours later.

3:00 P. M., head on reservoir gauge	12.73	feet.
5:00 P. M., " " " "	12 7092	"
Reduction of head in two hours	.02083	"

From this data and the reservoir measurements, above given, the leakage is estimated as 631.349 cubic feet for two hours, or at the rate of 2361.25 gallons per hour at observed head.

The duration of the capacity trials was fixed at five (5) hours for each engine, during which time all water pumped was delivered into the reservoir.

The capacity trial of engine No. 1 began at 12:17 A. M., January 10th, and terminated at 5:17 A. M., same date, with the following results:

Engine Counter at 12:17 A. M	93624
" " " 5:17 A. M	101358
Revolutions in 5 hours	7734

and piston speed

$$\frac{7734}{50} = 154.68 \text{ feet per minute.}$$

Water pressure gauge.

Minimum reading, corrected	56.15
Maximum " "	61.65
Mean of eleven readings	58.55

Data from reservoir.

Head in reservoir, at 12:17 A. M	8.5	feet.
" " " " 5:17 A. M	12.917	"
Head added, in five hours	4.417	"

The surface area of the reservoir at head of 8.5 feet, computed from data, is 25,967.735 square feet, at head of 12.917 feet is 30,249.893 square feet, and midway between these heads is 27,961.022 square feet.

Then by prismoidal formula the water added to reservoir was

$$\frac{(27{,}961.022 \times 4) + 25{,}967.735 + 30{,}249.893 \times 4.417 \times 7.48}{6} = 925{,}436.32 \text{ galls.}$$

To which must be added the leakage of reservoir for a period of five (5) hours, or

$$\frac{2{,}361.25 \times \sqrt{10.708} \times 5}{\sqrt{12.72}} = 10{,}832.35$$

gallons, making a total delivery into reservoir during capacity trial of engine No. 1 of 936,268.67 gallons.

Of this quantity a portion was the excess of injection water pumped into the reservoir.

The condensers furnished with the engines receive their injection water from the reservoir, the supply for which is raised from the pump well, or main suction pipe by a double acting piston pump (one to each engine) worked by a lever from one of the main pump rods.

The injection pumps are required to raise the water from the level in Wolf river, to the reservoir against a head (during the capacity trials) of fifty (50) feet, from which source the injection is drawn by gravity.

The capacity of the injection pumps is considerably in excess of the requirements of the condensers, and a certain surplus of water was in this manner delivered in the reservoir during the capacity trials, which has been estimated as follows:

Each injection pump has a diameter of 9 inches, and a stroke of 12.75 inches, with a rod (probably) 1.5 inch diameter, and allowing a moderate loss of action, delivered 6.58 gallons per revolution, or 50,889.72 gallons during capacity trial of engine No. 1; of this quantity from estimate based upon the known economy of engine, 37,847.08

gallons were absorbed by the condenser, leaving 13,042 64 gallons in the reservoir, from which is deduced the net delivery of main pumps for a period of five (5) hours as 924,226.03 gallons corresponding to a daily delivery under the terms of contract of

4,431,484.94 gallons.

The calculated delivery of two (2) pumps per revolution is 122.34 gallons and for five (5) hours

$$122.34 \times 7734 = 946,177.56 \text{ gallons,}$$

from which the loss of action is deduced, as

$$100 - \frac{923,226.03 \times 100}{946,177.56} = 2.43 \text{ per cent.}$$

(The pumps received water under a head of twelve (12) feet.)

The capacity trial of engine No. 2 commenced at 12:05 A. M., January 11th, and terminated at 5:05 A. M., same date, with the following results:

Engine Counter at 12:05 A. M	149971
" " " 5:05 A. M	157722
Revolutions in five (5) hours	7751

and piston speed

$$\frac{7751}{50} = 155.02 \text{ feet per minute.}$$

Water pressure gauge.

Minimum reading, corrected	56.15
Maximum " "	59.15
Mean of eleven readings	57.075

Data from reservoir.

Head in reservoir at 12:05 A. M	8.2708 feet.
" " " " 5:05 A. M	12.7083 "
Head added in five (5) hours	4.4375 "

The surface area of reservoir at head of 8.2708 feet computed from data is 25,742.537 square feet, at head of 12 7083 feet is 30,043.361 square feet, and midway between these heads is 27,865.848 square feet. Then by prismoidal formula the water added to reservoir, was

$$\frac{(27,865.848 \times 4) + 25,742.537 + 30,043.361 \times 4.4375 \times 7.48}{6} = 927,480.95 \text{ galls.}$$

To which is added the leakage of reservoir for a period of five (5) hours, or

$$\frac{2,361.25 \times \sqrt{10.49} \times 5}{\sqrt{12.72}} = 10,721.5$$

gallons, making a total delivery into reservoir during capacity trial of engine No. 2, of 938,202.45 gallons.

Of this quantity a portion was the surplus of injection, as before. Estimating net delivery of injection pump per revolution at 6.58 gallons, or 51,001 58 gallons during capacity test of engine No. 2, and computing from economy of engine as before—37,930 29 gallons absorbed by the condenser, then surplus of injection water pumped into reservoir was 13,071.29 gallons, and net delivery of main pumps for a period of five (5) hours was 925,131.16 gallons; corresponding to a daily delivery under terms of contract 4,440,629 57 gallons.

The calculated discharge for the five hours capacity trial of engine No. 2 is 122.34 × 7,751 = 948,257.34 gallons from which the loss of action is deduced as

$$100 - \frac{925,131.16}{948,257.34} \times 100 = 2.44 \text{ per cent.}$$

The close approximation of the slip in the trials for capacity, based upon independent measurements of water delivered, justifies the belief previously expressed, that the plungers of engine No. 2 were sensibly of the same diameters as the plungers of engine No. 1, which latter were carefully measured after the duty trials.

WORTHINGTON COMPOUND PUMPING ENGINE.

(At Buffalo, N. Y.)

The test for capacity involved an actual measurement of the water delivered by the pumps, for which two feasible methods offered.

The first by lowering the level of Prospect resevoir—closing all outlets—and pumping in a known volume of water, against an artificial head of 70 pounds on pump gauge produced by throttling with a 36″ stop valve in the discharge main; and the second, by making a special connection with the force main, at a point three miles from the pump house and diverting the delivery over a weir.

By the first method, the actual delivery of water upon which to estimate the capacity of pumps was necessarily small; and by the second method upwards of one hundred stop valves required closing for the period of weir measurements, with no means of estimating the probable leakage; besides, depriving a large section of the city of water during the hours of trial.

The first method had the advantage of time, in that the supply of water to all parts of the city might be made under direct pressure, while the reservoir was in use for test purposes.

After carefully canvassing both methods, it was finally decided to adopt the first, filling into the reservoir through such a section as was susceptible of reasonably accurate measurements.

In order that this method might be successfully employed, careful experiments (before and after the test for capacity), were made, to determine the tightness of walls and stop valves with no apparent leakage, and repeated measurements of lengths and slopes were made to insure correctness of the data upon which to estimate the discharge; the vertical rise or surface levels in the reservoir, were read from a measured rod divided in feet and tenths, with intermediate graduations to twentieths, which was carefully fixed and leveled in the South basin near the division wall. That portion of the reservoir above the division wall was selected for the test as offering the best facilities for close measurement, and the measured rod was so located that the arbitrary zero level of the water corresponded with 2.35 on the rod. The maximum rise of water level was agreed upon at 5 feet corresponding to 7.35 on the rod.

During the capacity trial when the surface of water in the reservoir coincided with the lowest and highest marks on the rod, the times were read to seconds from an accurate watch, and between these points the levels were read from the rod at the expiration of each regular quarter hour.

In order that the readings of counters in the pump house might agree for time with the readings of the measured rod in the reservoir, the rise of level in the latter was carefully noted, and a few minutes previous to the coincidence of the surface of water and the arbitrary zero point (2.35) on the rod, a messenger was dispatched from the reservoir to the pump house, upon whose arrival the assistants detailed for the purpose began minute readings of the engine counters. Directly the time was read for the agreement of water level with the zero point on the measured rod in the reservoir, a second messenger with a memorandum of the time, started for the pump house. Upon arrival of the second messenger from the reservoir the minute readings of the counters were discontinued, and readings of the instruments at the expiration of each regular quarter hour were substituted for the remainder of the trial. The same procedure was observed for the completion of trial. In this manner, with an agreement of time pieces at the two points of observation (reservoir and pump house), the reading of the engine counters at the time when the surface of the water coincided with any known point on the measured rod, can be read directly or interpolated from the record.

To insure corrections in the record, all data were taken by two intelligent observers, and all measurements were carefully repeated.

Two observers independently read the measured rod in the reservoir and agreed upon the readings; two more read the engine counters at the pump house, whilst the indications of the pressure gauges (steam and water) and the strokes (length) of plungers were ob-

served by the writer in behalf of the Water Board and by Mr. Johnson for the contractor.

The trial for capacity began at 4:57:30 P. M., July 2d, previous to which time the engine had been delivering into the reservoir for several hours, and terminated at 10:42:38 P. M. same date, embracing a period of 5 hrs. 45 min. 08 sec., during which interval the surface level of the reservoir was raised from 2.35 to 7.35 on the measured rod, or 5 feet head was added.

The section of reservoir filled was a true prismoid, of which the dimensions are given in the following table of reservoir measurements:

DIMENSIONS OF RESERVOIR.

Head 2.35 in gauge stick = 0 (feet) level.

Mean length $\dfrac{506.35 + 507.55}{2} = 506.95$ feet.

Mean width $\dfrac{175.7 + 174.5}{2} = 175.10$ feet.

Area $506.95 \times 175.1 = 88{,}766.945$ sq. ft.

Head 4.85 on gauge stick = 2.5 (feet) level.

Mean length $\dfrac{506.95 + 522.425}{2} = 514.6875$ feet.

Mean width $\dfrac{175.1 + 190.925}{2} = 183.0125$ feet.

Area $514.6875 \times 183.0125 = 94{,}194.2461$ sq. ft.

Head on gauge stick = 5 (feet) level.

Mean length $\dfrac{521 + 523.85}{2} = 522.425$ feet.

Mean width $\dfrac{191.35 + 190.5}{2} = 190.925$ feet.

Area $522.425 \times 190.925 = 99{,}743.993$ sq. ft.

Head added = 5 feet.

Then by prismoidal formula the volume of the section of reservoir filled represented

$$\frac{(94{,}194.2461 \times 4) + 88{,}766.945 + 99{,}743.993 \times 5 \times 7.48}{6} = 3{,}523{,}628.04838 \text{ U. S.}$$

standard gallons.

corresponding to a daily (24 hours) delivery at observed piston speed (93.772 feet) of

$$\frac{3,523,628.048 \times 86,400}{20,708} = 14,701,635.3 \text{ gallons.}$$

And at contract piston speed (110 feet), for which the boilers are at present entirely inadequate in heating and grate surface.

$$\frac{14,701,635.3 \times 110}{93.772} = 17,246,938.13 \text{ gallons.}$$

The counter reading at 4:57 P. M. July 2d, was 18,728 and at 4:58 P. M. same date 18,738, and by interpolation at 4:57:30 P. M. was 18,733.

The counter reading at 10:42 P. M. July 2d, was 22,630 and at 10:43 P. M. same date 22,642, and by interpolation at 10:42:38 was 22,637.6, from which the double strokes of one engine or quadruple strokes both engines, were,

$$22,637.6 - 18,733 = 3,904.6$$

The mean length of stroke engine No. 1 was 49.8125 inches, and mean length of stroke engine No. 2, was 49.651 inches, from which the mean piston speed during capacity trial, was,

$$\frac{49.8125 + 49.651 \times 3904.6}{12 \times 345.133} = 93.772 \text{ feet per minute.}$$

The calculated delivery of the pumps during the capacity trial has been estimated for the pump of engine No. 1, as

$$\frac{(38.1212^2 \times .7854) + (38.1212^2 \times .7854 - 5^2 \times .7854) \times 49.8125}{2 \times 231} = 244.005$$

U. S. standard gallons per single stroke.

For the pumps of engine No. 2, as

$$\frac{(38.1014^2 \times .7854) + (38.1014^2 \times .7854 - 5^2 \times .7854) \times 49.651}{2 \times 231} = 242.959$$

U. S. standard gallons per single stroke.

And a mean per single stroke for both pumps of

$$\frac{244.005 + 242.959}{2} = 243.482 \text{ gallons.}$$

And $243.482 \times 3,904.6 \times 4 = 3,802,799\ 2688$ U. S. standard gallons as the pump displacement, corresponding to a delivery of 3,523,628.048 gallons into the reservoir. From which the slip or loss of action of pumps is obtained, as

$$1 - \frac{3,523,628.048}{3,802,799.269} \times 100 = 7.34 \text{ per cent of calculated delivery.}$$

PRINCIPAL DIMENSIONS OF LONDON PUMPING ENGINES.

(At Main Pumping Stations.—Excepting Kent Works.)

Kirkwood.

Pumping Stations.	Engine.	Steam Cylinder. Diam....	Steam Cylinder. Stroke....
East London, Lea Bridge.....	Single acting, beam.......	100″	11′
" "	" "	84″	.....
" Old Ford......	" "	85″	10′
" "	" "	80″	10′
" "	" "	72″	10′
" "	" "	90″	11′
Southwark and Vauxhall, Hampton.	" bull..........	70″	10′
" "	" "	66″	10′
" "	" "	60″	10′
" "	" "	70″	10′
Grand Junction, Hampton ..	" "	60″	10′
" "	" "	60″	10′
" Kew........	" "	70″	10′
Gd. Junct., Camden Hill......	" "	70″	10′
" "	" "	70″	10′
W. Middlesex, Hampton.....	" "	64″	10′
" "	" "	64″	10′
Chelsea, Thames Ditton......	Rotative compound, two engines coupled. ..	28″ 46″	5.5′ 8.0′
" "	"	28″ 46″	5.5′ 8 0′
" "	"	28″ 46″	5.5′ 8.0′
Lambeth, "	"	28″ 46″	5.5′ 8.0′
" "	"	28″ 46″	5 5′ 8.0′
" "	"	28″ 46″	5.5′ 8.0′
New River Stoke, Newington................	"	28″ 46″	5.5′ 8.0′
"	"	28″ 46″	5.5′ 8.0′
"	Rotative sing. cylinder, 2 engs. coupled.	60″	8.0′

PRINCIPAL DIMENSIONS OF LONDON PUMPING ENGINES.

(At Main Pumping Stations.—Excepting Kent Works.)

Kirkwood.

Pump.	Water Cylinder.		Strokes per minute.....	Pumping head, pounds per sq. in......
	Diam....	Stroke...		
Plunger.	50"	11'	7 to 8	41.16
....	....	....	....	....
"	43"	9'	8 to 9	....
"	41"	9'	8	41.16
"	36"	10'	8 to 9	36.82
"	44"	11'	8.5	37.26
"	42"	10'	10	56.32
"	39"	10'	10	56.32
"	35"	10'	....	....
"	33"	10'	8 to 9	71.49
"	42"	10'	14	39.42
"	42"	10'	14	39.42
"	28"	10'	10 to 11	85.82
"	33"	10'	10	43.33
"	33"	10'	10	43.33
"	45"	10'	6.5	28.16
"	45"	10'	6.5	28.16
Bucket and Plunger.	24" 17.5"	7.1'	12 to 14	95.32
"	24" 17.5"	7.1'	12 to 14	95.32
"	24" 17.5"	7.1'	12 to 14	95.32
"	24" 17.5"	7.1'	13 to 15	83.32
"	24" 17.5"	7.1'	13 to 15	83.32
"	24" 17.5"	7.1'	13 to 15	83.32
"	27" 20"	6.92'	14	58.49
"	27" 20"	6.92'	14	58.49
Two buckets and plunger to each.	31.5" 22" 43" 30.5"	7' 6.75'	14 to 14.5	26 37.7

PERFORMANCE OF PUMPING ENGINES.

CORNISH BEAM ENGINES.

Location.	Date.	Engine.
United Mines........ ..	1842	Single cylinder, jacketed..................
Carn Brea.	1841	Compound, jacketed....... ....
Haarlem Meer.......	1848	" " ....
Cleveland, O..........	1873	Single cylinder...........................
Jersey City, N. J....	1856	" " ....
Louisville, Ky.......	1873	" " ....

CORNISH BULL ENGINE.

Cincinnati, O..	1872	Single cylinder, vertical . .. .

COMPOUND DUPLEX DIRECT ACTING.

Newark, N. J..........	1870	Horizontal, four cylinders...............
Philadelphia..........	1872	" " " ....
Charlestown, Mass....	1872	" " " ....
Toronto, Can..........	1872	" " " ....
Providence, R. I	1874	" " " ....
Toledo, O.....	1875	2 engines, horizontal, four cylinders.....
Lowell, Mass..........	1876	" " " " .....
Fall River, Mass......	1876	" " " " .....
Buffalo, N. Y.........	1882	Horizontal, four cylinders, jacketed......
Peoria. Ills...........	1882	" " " ... ...
Cleveland, O..........	1875	" " " 2 annular ...

COMPOUND ISOCHRONAL DIRECT ACTING.

Milwaukee, Wis.......	1878	Horizontal, two cylinders................
Cincinnati, O..........	1879	Vertical, " " ....
Springfield, O.........	1882	Horizontal, " " ....

COMPOUND CRANK AND FLY WHEEL.

Providence, R. I.....	1876	Vertical, two cylinders....... ....
Evansville, Ind..	1881	" " " jacketed..........
" "	1881	" " " " .. ..
Memphis, Tenn......	1882	" " " jacketed..........
" "	1882	" " " " ....

PERFORMANCE OF PUMPING ENGINES.

CORNISH BEAM ENGINES.

Designer.	Duty.†	Capacity.‡	Authority.
Taylor.	114,361,700*	....	Wm. Pole.
James Sims	101,702,000*	....	"
Gibbs & Dean.........	80,000,000	200,000,000	Appleton's Dict.
Allaire Works..... ...	41,774,955	5,711,988	Jour. Am. Soc. Civil Eng'ers.
West Point Foundry .	72,115,396	10,000,000	Copeland & Worthen.
T. R. Scowden.......	37,536,730*	3.816.575	Jour. Am. Soc. Civil Eng'ers.

CORNISH BULL ENGINE.

Geo. Shield..	23,580,687	11,847,481	Chas. Hermany.

COMPOUND DUPLEX DIRECT ACTING.

H. R. Worthington....	76,386.262	5,034,309	Geo. H. Bailey.
"	63,120,707	5,573,853	B'd of Experts.
"	56,937,643	5,000,000	Jour. Am. Soc. Civil Eng'ers.
"	63,561,306	12,000,000	Worthington.
" . ..	53,528,210	5,000,000	Smith, Graff & Reynolds.
" ...	45,611,924*	Each eng 2,800,000	Annual Report.
"	69.000,438	5,503,373	G. E. Evans.
"	70,977,177	5,500,000	Worthington.
"	67.812,170 61,968,284	17,247,000	John W. Hill.
"	24,573,664	2,000,000	"
W. M. Henderson. ...	31.968,006*	8,400.000	Annual Report.

COMPOUND ISOCHRONAL DIRECT ACTING.

Cope & Maxwell.... .	50,074,876	778,186	Hilbert & Reynolds.
"	Trial No. 1 53,957,957	2,258,986	Hill, Moore, & Ahrens.
	Trial No. 2 51,675,823	2,116,907	
"	53,592,518	3,089,518	C. A. Bauer.

COMPOUND CRANK AND FLY WHEEL.

A. F. Nagle...........	84,637,245	2,000,000	Hermany, Francis & Whitaker.
H. F. Gaskill.........	Engine No. 1. 81,885,917	3 872,101	John W. Hill.
"	Engine No. 2. 88,688.866	3,874,628	"
"	Engine No. 1. 99,672.837	4,431,485	"
"	Engine No. 2. 97,409,642	4,440,629	"

PERFORMANCE OF PUMPING ENGINES.

COMPOUND BEAM, CRANK AND FLY WHEEL.

Location	Date	Engine.
Lynn, Mass.	1873	Two cylinders, inclined, jacketed
Lawrence, Mass.	1876	jacketed, 2 engines
Lowell, Mass	1875	Vertical, two cylinders, jacketed
Trenton, N. J.	1876	" " " "
Milwaukee, Wis.	1875	two cylinders, vertical, unjacketed, 2 engines
Chicago	1877	Vertical, two cylinders, unjacketed
"	1877	" " " "
"	1877	" " " "
Pawtucket, R. I.	1878	Horizontal, two cylinders, jacketed
Providence, R. I.	1882	" " " "
Saratoga, N. Y.	1882	" four " "

SINGLE CYLINDER, BEAM, CRANK AND FLY WHEEL.

Location	Date	Engine.
Brooklyn, L. I.	1860	Vertical, Engine No. 1
" "	1860	" " No. 2
" "	1860	" " No. 3
New Bedford, Mass.	1869	"
Chicago	1874	" 2 engines coupled, unjacketed

SINGLE CYLINDER, CRANK AND FLY WHEEL.

Location	Date	Engine.
Cincinnati, O.	1872	2 engines coupled, horizontal, unjacketed, non-condensing
"	1872	Vertical, Harkness, condensing
"	1872	" Powell, "
Marion, Ind.	1877	2 engines coupled, horizontal, Scotch yoke, condensing.

COMPOUND QUADRUPLEX CRANK AND FLY WHEEL.

Location	Date	Engine.
Troy, N. Y.	1880	Four cylinders, inclined, condensing
" "	1880	" " " "
Buffalo, N. Y	1879	" " " "

DUPLEX DIRECT ACTING.

Location	Date	Engine.
Peoria, Ills	1882	Horizontal two cylinders, non-condensing, non-expanding

RADIAL CRANK ENGINE.

Location	Date	Engine.
Providence, R. I.	1874	Horizontal, five cylinders

PERFORMANCE OF PUMPING ENGINES.

COMPOUND BEAM, CRANK AND FLY WHEEL.

Designer.	Duty.†	Capacity.‡	Authority,
E. D. Leavitt	103,923,215	4,938,528	B'd of Experts
"	96,201,900	Each eng 4,979,234	"
James Simpson.......	72,925,00(*	4,207,785	Annual Report.
Wm. Wright.	84,500,000	2,086,523	F. J. Slade.
R. W. Hamilton.......	76,955,720	Each eng 8,683,720	B'd of Experts.
Quintard Works......	West eng'e 99,083,300	W. eng. 16,160,470	"
"	East engi'e 96,066,800	East eng. 15,571,970	"
"	75,000,000	......	Theron Skeel.
Geo. H. Corliss........	133,522,000	2,500,000	B'd of Experts.
"	113,035,000	9,105,604	S M. Gray.
H. F. Gaskill	112,899,983	4,850,200	John W. Hill.

SINGLE CYLINDER, BEAM, CRANK AND FLY WHEEL.

Wm. Wright..........	60,798,200	15,000,000	Smith, Graff & Worthen.
"	61,903,700	15,000,000	"
Hubbard & Whittaker	68,387,200	15,000,000	Worthen & Copeland.
W. J. McAlpine.......	59,336,497	5,000,000	B'd of Experts.
D. C. Cregier	65,824,581	36,000,000	"

SINGLE CYLINDER, CRANK AND FLY WHEEL.

Shield...............	43,566,178	4,702,805	Chas. Hermany.
T. R. Scowden........	87,789,990	4,651,987	"
"	34,064,977	4,263,297	"
Dean Bros........... .	49,231,207	1,500,000	J. D. Cook.

COMPOUND QUADRUPLEX CRANK AND FLY WHEEL.

Holly & Gaskill.......	Engine No. 1. 72,812,116	5,578,279	D. M. Greene.
"	Engine No. 2. 84,959,846	6,393,325	"
"	86,176,315	6,502,000	R. H. Buel.

DUPLEX DIRECT ACTING.

H. R. Worthington....	16,011,331	2,000,000	John W. Hill.

RADIAL CRANK ENGINE.

Geo. H. Corliss.......	25,865,740	5,000,000	Smith, Graff & Reynolds.

* Said to be average duty, all others obtained by special trials.
† The Duty is stated in foot pounds of work per hundred lbs. of coal,
‡ Capacity is stated in gallons per day of 24 hours.

THE PROPERTIES OF WATER.

Water was supposed to be an element, until Priestly late in the eighteenth century, discovered that when hydrogen was burned in a glass tube, water was deposited on the sides.

The several conditions of water are usually stated as the solid, the liquid and the gaseous. Two conditions are covered by the last term, and water should be understood as capable of existing in four different conditions—the solid, the liquid, the vaporous. and the gaseous. At and below 32° Fahr. water exists in the solid state, and is known as ice. According to Prof. Rankine, ice at 32° has a specific gravity of .92. Thus a cubic foot of ice weighs 57.45 lbs.

When water passes from the solid to the liquid state, heat is required for liquefaction, sufficient to elevate the temperature of one pound of water 143° Fahr. This is termed the latent heat of liquefaction. According to M. Person, the specific heat of ice is .504, and the latent heat of liquefaction 142.65.

From 32° to 39° the density of water increases; above the latter temperature the density diminishes.

Water is said to be at its maximum density at 39° F.; and under pressure of one atmosphere weighs, according to Berzelius, 62.382 lbs. per cubic foot. The following formula may be used to estimate the weight of water at any other temperature.

Let D' = weight of water per cubic foot at temperature of maximum density (39 2 F.).

T = any temperature on Fahr. scale.

D = weight of water per cubic foot at temperature T.

Then—

$$D = \frac{2D'}{\dfrac{T + 461}{500} + \dfrac{500}{T + 461}}$$

Desired the weight of a cu. ft. of water at temperature 60° F.

$$D = \frac{62.382 \times 2}{\dfrac{60 + 461}{500} + \dfrac{500}{60 + 461}} = \frac{124.764}{2.0017} = 62.33$$

Water is said to vaporize at 212° Fahr, and pressure of one atmosphere (14.7 lbs:), but Faraday has shown that vaporization occurs at all temperatures from absolute zero, and that the limit to vaporization

is the disappearance of heat. Dalton obtained the following expermental results on evaporation below the boiling temperature:

Temp.	Rate of Evaporation.	Barometer.
212	1.00	29.92
180	.50	15.27
164	.33	10 59
152	.25	7.93
144	.20	6 488
138	.17	5.565

From this, the general law is deduced that the rate of surface evaporation is proportional to the elastic force of the vapor.

Thus, suppose two tanks of similar surface dimensions and open to the atmosphere, one containing water maintained constantly at 212° Fahr., and the other containing water at 144° Fahr.

Then for each pound of water evaporated in the last tank, five pounds will be evaporated in the first tank.

It should be understood that the law of Dalton holds good only for dry air, and when the air contains vapor having an elastic force, equal to that of the vapor of the water, the evaporation ceases.

The boiling point of water depends upon the pressure. Thus at one atmosphere (14.7 lbs. = 29 92″ barometer) the temperature of ebullition is 212° Fahr. With a partial vacuum, or absolute pressure of one pound (2.037″ of mercury) the boiling point is 101.40 Fahr.

Upon the other hand, if the pressure be 74.7 lbs. absolute (60 lbs. by gauge), the temperature of evaporation becomes 307° Fahr.

The relations of temperature and pressure have been made the subject of special investigation from the time of Watt, down to the celebrated experiments of Regnault, which have been accepted as conclusive so far as they extended.

The relations of pressure and density, however, have not been determined by experiment. Messrs. Fairbairn and Taie have investigated this problem and deduced a formula, but late experience has shown that while the Fairbairn and Tate formula is perhaps the best of its kind, it can not be accepted as correctly stating the relations of pressure and density. (Density of saturated steam, Van Nostrand's Magazine, June, 1878.)

The vaporous condition of water is limited to saturation. That is to say, when water has been converted by heat into vapor (steam), and when this vapor has been furnished with latent heat sufficient to render it anhydrous, the vaporous condition ends, and the gaseous state begins. Superheated steam is water in the gaseous state.

The temperature of the gaseous state of water, like that of the vaporous, depends upon the imposed pressure. Under pressure of one atmosphere, water exists in the solid state at and below 32° Fahr.; from 32° to 212° it exists in the liquid state; at and above 212° in the vaporous state; and above saturation in the gaseous state.

It has been stated that water boils at 212°, but M. M. Magnus and Donney have shown that, when water is freed of air, it may be elevated in temperature to 270° before evaporation takes place.

The specific heat of water under the several conditions are as follows:

Solid	.504
Liquid, at 39. 2 F,	1.000
Vaporous	.475 to 1.000
Gaseous	.475

HYDRAULIC FORMULAE.

Velocity is usually stated in feet per second, and is first calculated as for a body falling freely in vacuo, and then modified by a proper co-efficient according to the conditions subsisting in any particular case,

$$v = \sqrt{h\,2\,g} \quad \text{or } 8.025\sqrt{h}$$

Where h = head, and g = acceleration of gravity = 32.2.

Conversely the head due any given velocity is

$$h = \frac{v^2}{2\,g}$$

All matter in motion develops a frictional resistance the value of which, in terms of the head, must be added to the head due velocity to state a true or total head.

Suppose a delivery of 4,302,069.1 gallons of water per diem through a 24″ pipe, 410 feet long, laid horizontally. The discharge per second would be 6.65675 cubic feet. The area of such a pipe is 3.1416 square feet, and the velocity of flow $\frac{6.65675}{3.1416} = 2.1189$ feet, corresponding to a head of $\frac{2.1189^2}{64.4} = .0697$ feet.

And the frictional resistance by the generally adopted Weisbach formula $F = \frac{v^2}{2\,g} \times \frac{L}{d} \times \left(.0144 + \frac{.01716}{\sqrt{v}}\right)$

Where F = friction head in feet, L = length of pipe in feet, and d = diameter of pipe in feet; whence—

$$\frac{2.1189^2}{64.4} \times \frac{410}{2} \times \left(.0144 + \frac{.01716}{\sqrt{2.1189}}\right) = .37428 \text{ foot,}$$

and true head .0697 + .37428 = .44398 foot.

Hawksley, gives a formula for the discharge of water through straight pipes free from incrustation and bends, as follows:

$$v = 48\sqrt{\frac{h\,d}{L}}$$

When h = head in feet; d = diameter of pipe in feet; and L, length of pipe in feet, applying this method to above head, length, and diameter of pipe, the velocity would be,

$$v = 48\sqrt{\frac{.44398 \times 2}{410}} = 2.14038 \text{ feet.}$$

Mr. Simpson, also gives a formula for the flow of water through straight cylindrical pipes, as follows:

$$v = 50\sqrt{\frac{h \times d}{L + 50\,d}}$$

Applying which to above data the velocity becomes,

$$v = 50\sqrt{\frac{.44398 \times 2}{410 + (50 \times 2)}} = 2.0865 \text{ feet.}$$

Both the latter formulas take cognizance of the frictional resistance of the sides of the pipe and are intended to give the actual velocity of flow.

In view of the fact that water pipes are seldom straight, seldom of uniform section from end, and, seldom free from incrustations, or other obstructions, it is preferable in the author's opinion, to employ the Simpson formula, which as will be observed recognizes a greater loss of head by friction, and produces a lower velocity of flow.

The formula quoted from Weisbach, is true only for a straight, smooth pipe, and will always produce a friction head less than the true head, which discrepancy may be accounted for by extra frictional resistances in the pipe, not considered by the formula.

To illustrate this, the engines furnishing the public water supply at Evansville, Indiana, draw from the Ohio river through a suction pipe consisting of 200 feet of 16-inch pipe, 1,300 feet of 16-inch pipe, and 410 feet of 24-inch pipe, with 2-16-inch elbows, and 3-24-inch elbows.

The estimated friction head for a daily delivery of 4,000,000 gallons is 1.85586 feet, while the actual head as measured was 2.5925 feet.

RESISTANCE OF CIRCULAR BENDS.

Weisbach, from his own experiments and those of Du Buat, proposed the following formulæ for the frictional resistance of curved bends or elbows in lines of pipe:

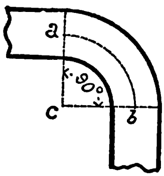
Let R = radius of curve or bend, in inches or feet.
r = radius of section of pipe, in inches or feet.
K = co-efficient of resistance.

Then—

$$K = 0.131 + 1.847 \left(\frac{r}{R}\right)^{\frac{7}{2}}$$ for pipes of circular cross section.

And—

$$K = 0.124 + 3.104 \left(\frac{r}{R}\right)^{\frac{7}{2}}$$ for pipes of rectangular cross section.



Let v = velocity of flow, in feet per second.
a° = angle embraced by curve or bend.
(a right angle bend = 90°.)
h = friction head in feet for bend.

Then—

$$h = K \cdot \frac{v^2}{2g} \cdot \frac{a^\circ}{18^\circ}$$

Let $n = \frac{r}{R}$ and K = corresponding co-efficient of resistance, then the following tables for bends of circular and rectangular cross sections, computed by above formulæ, contain the values of n and K for ratios of 0.1 to 1.0:

Bends of Circular Cross Sect.				Bends of Rectang'r Cross Sect.			
$K = 0.131 + 1.847 \left(\frac{r}{R}\right)^{\frac{7}{2}}$				$K = 0.124 + 3.104 \left(\frac{r}{R}\right)^{\frac{7}{2}}$			
$n = \frac{r}{R}$	K	$n = \frac{r}{R}$	K	$n = \frac{r}{R}$	K	$n = \frac{r}{R}$	K
0.10	0.131	0.60	0.440	0.10	0.124	0.60	0.644
0.15	0.135	0.65	0.540	0.15	0.128	0.65	0.811
0.20	0.138	0.70	0.661	0.20	0.135	0.70	1.015
0.25	0.150	0.75	0.800	0.25	0.148	0.75	1.258
0.30	0.158	0.80	0.977	0.30	0.170	0.80	1.545
0.35	0.180	0.85	0.180	0.35	0.203	0.85	1.881
0.40	0.206	0.90	0.408	0.40	0.249	0.90	2.271
0.45	0.240	0.95	0.680	0.45	0.313	0.95	2.718
0.50	0.294	1.00	0.978	0.50	0.398	1.00	3.228
0.55	0.350			0.55	0.507		

What head is required to overcome the friction for a 90° bend or elbow, the diameter of which is 20 inches, and the radius of curvature 25 inches, with a velocity of flow of 2.7896 feet per second.

$r = 10$ inches, $R = 25$ inches, and

$$\frac{r}{R} = n = .4$$

And K, from table of co-efficients for bends of circular cross section, corresponding to a ratio $n = .4$ is .206.

Then—

$$h = \frac{2.7896^2 \times 90}{64.4 \times 180} \times .206 = .01245 \text{ foot.}$$

Suppose the section of above elbow is square, what then would be the friction head?

$r =$ (as before) 10 inches, $R = 25$ inches.

$\frac{r}{R} = n = .4$, the co-efficient of which is $K = .249$.

Then—

$$h = \frac{2.7896^2 \times 90}{64.4 \times 180} \times .249 = .01504 \text{ feet.}$$

The following table for frictional resistance of bends has been calculated by Mr. Trautwine with the Weisbach formula—

$$h = K \frac{v^2}{2g} \frac{a^\circ}{180}$$

HEADS REQUIRED TO OVERCOME THE RESISTANCE OF 90 DEG. CIRCULAR BENDS.

RADIUS OF BEND IN DIAMS. OF PIPE.

Velocity in feet Per sec.	0.5	0.75	1.00	1.25	1.5	2.0	3.0	5.0
	Head, in feet.							
1	.016	.005	.002	.002	.001	001	.001	001
2	.062	.018	.009	007	005	005	.004	004
3	.140	.041	.020	.015	.012	.011	.010	009
4	.248	.072	.036	.026	.021	.019	017	016
5	.388	.113	.056	.041	.033	029	027	025
6	.559	.162	.081	.059	048	042	038	036
7	.761	.221	.110	.080	.066	057	.052	050
8	.994	.288	.144	.104	.086	074	.069	065
9	1.260	.365	.182	.132	.108	.094	.086	082
10	1.550	.450	.225	.163	.134	.116	.106	101
12	2.240	.649	.324	.235	.192	.167	.153	.145

DISCHARGE OF LONG IRON PIPES.

Let H = head, or vertical distance from center of inlet to center of outlet, in feet.
L = length of pipe, in feet.
D = diameter of pipe, in feet.
f = co-efficient for frictional resistance of surface of pipe.
A = area of pipe, in sq. feet.
p = wetted perimeter of pipe, in feet.
m = hydraulic mean depth, $= \frac{A}{p} = \frac{D}{4}$
v = velocity, in feet per second.
Q = discharge in cubic feet per second =

$$\left(\frac{\text{discharge in U. S. standard gallons}}{7.48}\right)$$

(According to Darcy.)

$$f = .005\left(1 + \frac{1}{48\,m}\right) = .005\left(1 + \frac{1}{12\,D}\right) \text{ for round pipes.}$$

Then—

$$v = 8.025\sqrt{\frac{H\,D}{4\,f\,L}} = 53\sqrt{\frac{H\,D}{L}} \text{ nearly,}$$

and—

$$Q = v\,A = 6.303\sqrt{\frac{H}{4\,f\,L}}\cdot\sqrt{D^5}$$

Let $H = 45$ feet. $L = 11{,}391$ feet. $D = 7'' = \frac{7}{12} = .5833$ feet.

$$4\,f = .02\left(1 + \frac{1}{12 \times .5833}\right) = .02\left(1 + \frac{1}{7}\right) = .02286$$

and—

$$v = 8.025\sqrt{\frac{45 \times .5833}{.02286 \times 11{,}391}} = 2.5478 \text{ feet,}$$

and—

$$Q = 2.5478\,A = .68084 \text{ cu. ft.} = .68084 \times 60 \times 7.48 = 305.56$$

gallons per minute, and by second equation—

$$Q = 6.303\sqrt{\frac{45}{.02286 \times 11{,}391}} \times \sqrt{.5833^5} = .68087 \text{ cu. ft.}$$

Again—

$$H = \frac{4\,f\,L}{D}\,\frac{v^2}{2\,g} = \frac{.02282 \times 11{,}391}{.5833} \times \frac{2.5478^2}{64.4} = 45 \text{ ft.}$$

For rough approximation, Rankine suggests that $4\,f$ may be taken as 0258, which is to be used in cases where the discharge, Q, = length, L, and head, H, are given, and the diameter, D, is desired

Then—

$$D = \sqrt[5]{\frac{4\,f\,L\,Q^2}{39.73\,H}}$$

But f depends upon D, and D is unknown; hence D must be obtained by a tentative process, for which Rankine proposes the following formulæ:

Let D' = approximation of D.

f' = one approximation of f = .00645.

f'' = another approximation of f.

Then—

$$D' = .2306\,\sqrt[5]{\frac{L\,Q^2}{H}}$$

and—

$$f'' = .005\left(1 + \frac{1}{12\,D'}\right)$$

and, finally,

$$D = D'\sqrt[5]{\frac{f''}{f'}} = D'\sqrt[5]{\frac{f''}{.00645}}$$

Suppose, as before, $Q = .68087$ cubic feet, $L = 11{,}391$ feet, and $H =$ 45 feet; desired D.

Then—

$$D' = .2306\,\sqrt[5]{\frac{11{,}391 \times .68087^2}{45}} = .598 \text{ foot,}$$

and—

$$f'' = .005\left(1 + \frac{1}{12 \times .598}\right) = .005696$$

and—

$$D = .598\,\sqrt[5]{\frac{.005696}{.00645}} = .5834 \text{ foot.}$$

The following table of fifth powers and roots may be used for approximations; but for accuracy in estimating the discharge of pipes above formula should be worked with logarithms.

TABLE OF FIFTH ROOTS AND FIFTH POWERS.

Trautwine.

Power.	No. or Root.	Power.	No. or Root.	Power.	No. or Root.
.0000100	.1	.001721	.280	.135012	.67
.0000110	.102	.001880	.285	.145393	.68
.0000122	.104	.002051	.290	.156403	.69
.0000134	.106	.002234	.295	.168070	.70
.0000147	.108	.002430	.300	.180423	.71
.0000161	.110	.002639	.305	.193492	.72
.0000176	.112	.002863	.310	.207307	.73
.0000193	.114	.003101	.315	.221901	.74
.0000210	.116	.003355	.320	.237405	.75
.0000229	.118	.003626	.325	.253553	.76
.0000249	.120	.003914	.330	.270678	.77
.0000270	.122	.004219	.335	.288717	.78
.0000293	.124	.004544	.340	.307706	.79
.0000318	.126	.004888	.345	.327680	.80
.0000344	.128	.005252	.350	.348678	.81
.0000371	.130	.005638	.355	.370740	.82
.0000401	.132	.006047	.360	.393904	.83
.0000432	.134	.006478	.365	.418212	.84
.0000465	.136	.006934	.370	.443705	.85
.0000500	.138	.007416	.375	.470427	.86
.0000538	.140	.007924	.380	.498421	.87
.0000577	.142	.008459	.385	.527732	.88
.0000619	.144	.009022	.390	.558406	.89
.0000663	.146	.009616	.395	.590490	.90
.0000710	.148	.010240	.400	.624032	.91
.0000754	.150	.011586	.41	.659082	.92
.0000895	.155	.013069	.42	.695688	.93
.000105	.160	.014701	.43	.733904	.94
.000122	.165	.016492	.44	.773781	.95
.000142	.170	.018453	.45	.815373	.96
.000164	.175	.020596	.46	.858734	.97
.000189	.180	.022935	.47	.903921	.98
.000217	.185	.025480	.48	.950990	.99
.000248	.190	.028248	.49	1.	1.
.000282	.195	.031250	.50	1.10408	1.02
.000320	.200	.034503	.51	1.21665	1.04
.000362	.205	.038020	.52	1.33823	1.06
.000408	.210	.041820	.53	1.46933	1.08
.000459	.215	.045917	.54	1.61051	1.10
.000515	.220	.050328	.55	1.76234	1.12
.000577	.225	.055073	.56	1.92541	1.14
.000644	.230	.060169	.57	2.10034	1.16
.000717	.235	.065636	.58	2.28775	1.18
.000796	.240	.071492	.59	2.48832	1.20
.000883	.245	.077760	.60	2.70271	1.22
.000977	.250	.084460	.61	2.93163	1.24
.001078	.255	.091613	.62	3.17580	1.26
.001188	.260	.099244	.63	3.43597	1.28
.001307	.265	.107374	.64	3.71293	1.30
.001435	.270	.116029	.65	4.00746	1.32
.001573	.275	.125233	.66	4.32040	1.34

TABLE OF FIFTH ROOTS AND FIFTH POWERS.—Continued.

Power.	No. or Root.	Power.	No. or Root.	Power.	No. or Root.
4.65259	1.36	310.136	3.15	14539	6.80
5.00490	1.38	335.544	3.20	15640	6.90
5.37824	1.40	362.591	2.25	16807	7.00
5.77353	1.42	391.354	2.30	18042	7.10
6.19174	1.44	421.419	3.35	19349	7.20
6.68383	1.46	454.354	3.40	20731	7.30
7.10082	1.48	488.760	3.45	22190	7.40
7.59375	1.50	525.219	3.50	23730	7.50
8.11368	1.52	563.822	3.55	25355	7.60
8.66171	1.54	604.662	3.60	27068	7.70
9.23896	1.56	647.835	3.65	28872	7.80
9.84658	1.58	693.440	3.70	30771	7.90
10.4858	1.60	741.577	3.75	32768	8.00
11.1577	1.62	792.352	3.80	34868	8.10
11.8637	1.64	845.870	3.85	37074	8.20
12.6049	1.66	902.242	3.90	39390	8.30
13.3828	1.68	961.580	3.95	41821	8.40
14.1986	1.70	1024.00	4.00	44371	8.50
15.0537	1.72	1089.62	4.05	47043	8.60
15.9495	1.74	1158.56	4.10	49842	8.70
16.8874	1.76	1230.95	4.15	52773	8.80
17.8690	1.78	1306.91	4.20	55841	8.90
18.8957	1.80	1386.58	4.25	59049	9.00
19.9690	1.82	1470.08	4.30	62403	9.10
21.0906	1.84	1557.57	4.35	65908	9.20
22.2620	1.86	1649.16	4.40	69569	9.30
23.4849	1.88	1745.02	4.45	73390	9.40
24.7610	1.90	1845.28	4.50	77378	9.50
26.0919	1.92	1950.10	4.55	81537	9.60
27.4795	1.94	2059.63	4.60	85873	9.70
28.9255	1.96	2174.03	4.65	90392	9.80
30.4317	1.98	2293.45	4.70	95099	9.90
32.0000	2.00	2418.07	4.75	100000	10.0
36.2051	2.05	2548.04	4.80	110408	10.2
40.8410	2.10	2683.54	4.85	121665	10.4
45.9401	2.15	2824.75	4.90	133823	10.6
51.5363	2.20	2971.84	4.95	146933	10.8
57.6650	2.25	3125.00	5.00	161051	11.0
64.3634	2.30	3450.25	5.10	176234	11.2
71.6703	2.35	3802.04	5.20	192541	11.4
79.6262	2.40	4181.95	5.30	210034	11.6
88.2735	2.45	4591.65	5.40	228776	11.8
97.6562	2.50	5032.84	5.50	248832	12.0
107.820	2.55	5507.32	5.60	270271	12.2
118.814	2.60	6016.92	5.70	293163	12.4
130.686	2.65	6563.57	5.80	317580	12.6
143.489	2.70	7149.24	5.90	343597	12.8
157.276	2.75	7776.00	6.00	371293	13.0
172.104	2.80	8445.96	6.10	400746	13.2
188.029	2.85	9161.33	6.20	432040	13.4
205.111	2.90	9924.37	6.30	465259	13.6
223.414	2.95	10737	6.40	500490	13.8
243.000	3.00	11603	6.50	537824	14.0
263.936	3.05	12523	6.60	577353	14.2
286.292	3.10	13501	6.70	619174	14.4

TABLE OF FIFTH ROOTS AND FIFTH POWERS.—Concluded.

Power.	No. or Root	Power.	No. or Root.	Power.	No. or Root.
663383	14.6	11431377	25.8	241806543	47.5
710082	14.8	11881376	26.0	254803968	48.0
759375	15.0	12345437	26.2	268354383	48.5
811368	15.2	12823886	26.4	282475249	49.0
866171	15.4	13317055	26.6	299184391	49.5
923896	15.6	13825281	26.8	312500000	50.0
984658	15.8	14348907	27.0	345025251	51
1048576	16.0	14888280	27.2	380204032	52
1115771	16.2	15443752	27.4	418195493	53
1186367	16.4	16015681	27.6	459165024	54
1260493	16.6	16604430	27.8	503284375	55
1338278	16.8	17210368	28.0	550731776	56
1419857	17.0	17833868	28.2	601692057	57
1505366	17.2	18475309	28.4	656356768	58
1594947	17.4	19135075	28.6	714924299	59
1688742	17.6	19813557	28.8	777600000	60
1786899	17.8	20511149	29.0	844596301	61
1889568	18.0	21228253	29.2	916132832	62
1996903	18.2	21965275	29.4	992436543	63
2109061	18.4	22722628	29.6	1073741824	64
2226203	18.6	23500728	29.8	1160290625	65
2348493	18.8	24300000	30.0	1252332576	66
2476099	19.0	26393634	30.5	1350125107	67
2609193	19.2	28629151	31.0	1453933568	68
2747949	19.4	31013642	31.5	1564031349	69
2892547	19.6	33554432	32.0	1680700000	70
3043168	19.8	36259082	32.5	1804229351	71
3200000	20.0	39135393	33.0	1934917632	72
3363232	20.2	42191410	33.5	2073071593	73
3533059	20.4	45435424	34.0	2219006624	74
3709677	20.6	48875980	34.5	2373046875	75
3893289	20.8	52521875	35.0	2535525376	76
4084101	21.0	56382167	35.5	2706784157	77
4282322	21.2	60466176	36.0	2887174368	78
4488166	21.4	64783487	36.5	3077056399	79
4701850	21.6	69343957	37.0	3276800000	80
4923597	21.8	74157715	37.5	3486784401	81
5153632	22.0	79235168	38.0	3707398432	82
5392186	22.2	84587005	38.5	3939040643	83
5639493	22.4	90224199	39.0	4182119424	84
5895793	22.6	96158012	39.5	4437053125	85
6161327	22.8	102400000	40.0	4704270176	86
6436343	23.0	108962013	40.5	4984209207	87
6721093	23.2	115856201	41.0	5277319168	88
7015834	23.4	123095020	41.5	5584059449	89
7320825	23.6	130691232	42.0	5904900000	90
7636332	23.8	138657910	42.5	6240321451	91
7962624	24.0	147008443	43.0	6590815232	92
8299976	24.2	155756538	43.5	6956883693	93
8648666	24.4	164916224	44.0	7339040224	94
9008978	24.6	174501858	44.5	7737809375	95
9381200	24.8	184528125	45.0	8153726976	96
9765625	25.0	195010045	45.5	8587340257	97
10162550	25.2	205962976	46.0	9039207968	98
10572278	25.4	217402615	46.5	9509900499	99
10995116	25.6	229345007	47.0		

FLOW OF WATER IN OPEN CHANNELS.

The following formulæ assumes the chanuel to be straight, and of uniform transverse profile for a given length, L.

Let L = length, in feet, of channel.
A = area of cross section, in feet.
h = surface slope, in feet, for length, L.
p = wet perimeter, in feet.
v = velocity of flow, in feet, per second.
D = volume of flow, in cubic feet, per second.

Then—

$$v = 92.26 \sqrt{\frac{A\ h}{p\ L}}$$

$$h = .00011747 \frac{L\ p}{A} v^2 \qquad \text{or—} \qquad h = .007565 \frac{L\ p\ v^2}{A\ 2g}$$

and—

$$D = 92.26 \sqrt{\frac{A\ h}{p\ L}} \times A = A\ v$$

The trapezoidal profile is generally adopted for open water courses of earth work, and the rectangular and semicircular profiles are generally adopted for channels of wood, stone, or iron.

What volume of water will pass per second in a channel of trapezoidal section, the length, L, of which is 5,000 feet, the bottom width 10 feet, the surface width 26 feet, and the depth 6 feet, with a surface slope of 1 foot.

$$p = 10 + 2\sqrt{8^2 + 6^2} = 30 \text{ feet,}$$

$$A = 6\ \frac{10 + 26}{2} = 108 \text{ square feet,}$$

$$v = 92.26 \sqrt{\frac{108 \times 1}{30 \times 5{,}000}} = 2.475 \text{ feet, and}$$

$$D = 108 \times 2.475 = 267.3 \text{ cubic feet per second,}$$

$$h = .00011747 \left(\frac{5{,}000 \times 30}{108}\right) 2.475^2 = 1 \text{ foot.}$$

The co-efficient of friction, .00011747, deduced by Eytelwien from

experiments of Du Buat and others, must be corrected for the flow of water in rivers, and similar natural water courses, by the formula proposed by Weisbach, from his own and other experiments, as follows:

$$c = .007409 \left(1 + \frac{.1920}{v'}\right)$$

v' being determined approximately by formula—

$$v' = 92.26 \sqrt{\frac{A\ h}{p\ L}}$$

and v, or corrected velocity, by the formula—

$$v, = \sqrt{\frac{A}{c\ L\ p}\ 2\ g\ h}$$

Desired the volume of flow of a stream, with a width of 50 feet, mean depth of 6 feet, wetted perimeter of 60 feet, and fall of 6 inches (.5 foot) in 500 feet.

$$A = 50 \times 6 = 300 \text{ square feet.}$$

$$v' = 92.26 \sqrt{\frac{300 \times .5}{60 \times 500}} = 6.5237 \text{ feet.}$$

Then—

$$c = .007409 \left(1 + \frac{.1920}{6.5237}\right) = .007627 \text{ nearly,}$$

and—

$$v = \sqrt{\frac{300 \times 64.4 \times .5}{.007627 \times 500 \times 60}} = 6.498 \text{ feet and}$$

volume of flow.

$$D = 300 \times 6.498 = 1949.4 \text{ cubic feet per second.}$$

When the head is desired, the volume of flow, area of cross section, wetted perimeter and length being given. Reduce volume of flow to mean velocity, v.

Then—

$$h = c\,\frac{L\ p\ v^2}{A\ 2g} = .007409 \left(1 + \frac{.1920}{v}\right) \times \frac{L\ p\ v^2}{A\ 2\ g}$$

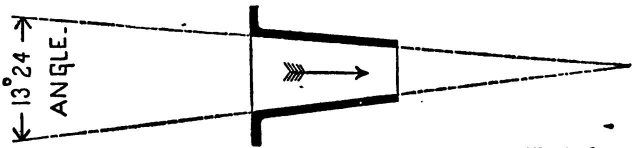
CO-EFFICIENTS OF EFFLUX AND VELOCITY.

For Conically Convergent Tubes or Mouth Pieces.

The following results, from experiments by d Aubuison and Castel

upon ajutages, were obtained from tubes, uniformly .6102 inch diameter at orifice of efflux, and 1.58652 inches long, operated under constant heads of 9.842125 feet.

The discharge was measured by a gauged vessel; and the range of jet corresponding to the constant head for each mouth piece was also measured to determine the co-efficients of efflux, of velocity, and of contraction.

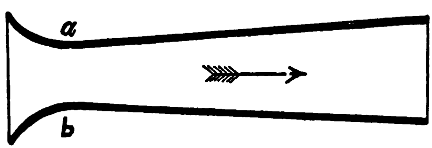


Weisbach.

Angle of Converg-ence.	Co-efficient of Efflux.	Co-efficient of Velocity.	Angle of Converg-ence.	Co-efficient of Efflux.	Co-efficient of Velocity.
0° 0′	0.829	0.829	13° 24′	0.946	0.963
1° 36′	0.866	0.867	14° 28′	0.941	0.966
3° 10′	0.895	0.894	16° 36′	0.938	0.971
4° 10′	0.912	0.910	19° 28′	0.924	0.970
5° 26′	0.924	0.919	21° 0′	0.919	0.972
7° 52′	0.930	0.932	23° 0′	0.914	0.974
8° 58′	0.934	0.942	29° 58′	0.895	0.975
10° 20′	0.938	0.951	40° 20′	0.870	0.980
12° 4′	0.942	0.955	48° 50′	0.847	0.984

The angle of convergence of first and fourth columns, is the angle enclosed by the projected walls of the mouth piece, or twice the angle enclosed by the projected side and axis of the tube.

In a convergent mouth piece, the orifice of efflux is at the smallest end.



Eytelwein, with a tube similar in form to the figure, the dimensions of which were 1.09356 inches diameter at the throat, and 1.959295 inches diameter at the orifice of efflux, and 8.8125 inches long, found

the discharge to be 2.5 times the discharge through thin plate with orifice a—b, and 1.9 times the discharge of a short cylindrical pipe of diameter a—b.

Jet, deau.—Box.

$H', = \frac{H^2}{8 \times d} \times .0125$, where H = head on jet in feet, d = diameter of pipe in inches. and H' = difference between height of Jet and, H

What heighth will a jet attain from a nozzle 1 inch diameter under a pressure of 150 pounds, $H = 150 \times 2.308 = 346.2$ feet, and $H' = \frac{346.2^2}{8} \times 0125 = 187.272$, and $H - H' = 346.2 - 187.272 = 158.928$ feet.

Again, suppose the head to be 130 pounds or 300 feet, then $H = \frac{300^2}{8} \times .0125 = 140.625$, and $H - H', = 300 - 140.625 = 159.375$ feet: from which it appears that with one inch nozzles, a head or pressure of 130 pounds will produce the maximum altitude of jet.

Discharge of Nozzles Adapted from Box Hydraulics.

$G = \sqrt{H} \times (8\,d)^2 \times .288$, when H = head in feet, d = diameter in inches, and G = U. S. standard gallons discharged per minute.

U. S. standard	gallon, 231	cu. inches.
British Imperial	" 277.274	"

Substituting .24 for .288 in above formula produces discharge in Imperial gallons.

What will be the discharge of a jet one and one-quarter inch diameter, under a pressure of 130 pounds?

$H = 130 \times 2.308 = 300.04$ feet, and $\sqrt{300.04 \times 10^2} \times .288 \doteq 498.8736$ U. S. gallons, or 415.73 Imperial gallons. Six 1.25 inch (diameter) fire streams will consume, under a pressure of 130 pounds, nearly 3,000 gallons per minute, or more than 4,000,000 gallons per diem.

Flow of Water over Weirs.

The well known Francis formula for discharge of weirs is

$Q = 3.33\,(L\ .1\,n\,H)\,H^{\frac{3}{2}}$ and the correction for velocity of approach $H' = [(H + h)^{\frac{3}{2}} - h^{\frac{3}{2}}]^{\frac{2}{3}}$ in which H = observed head on weir, in feet, n = number of end contractions, L = length of weir in feet.

h = head in feet due velocity of approach = apparent discharge (Q)

divided by cross section of stream, and H' = corrected head whence true discharge becomes

$$Q' = 3.33\,(L - .1\,n\,H')\,H'^{\frac{3}{2}}$$

The following data is from the expert trial of the Warden Compound pumping engine at the Hunt street station, Cincinnati, March, 1879.

$H = .4549$ foot, $L = 3.0013$ feet, $n = 2$, cross section of weir box $A = 3.1149 \times 4 = 12.4596$ feet, then

$$Q = 3.33 \times [3.0013 - (2 \times .4549)]\ .4549^{\frac{3}{2}} = 2.9734 \text{ cubic feet,}$$

$$v = \frac{2.9734}{12.4596} = .23864, \text{ and } h = \frac{.23864^2}{2\,g} = .00088 \text{ foot,}$$

$$\text{and corrected head } H' = [(.4549 + .00088)^{\frac{3}{2}} - .00088^{\frac{3}{2}}]^{\frac{2}{3}} = .45574,$$

and corrected discharge, $Q' = 3.33\ [3.0013 - (.2 \times .45574)]\ .45574^{\frac{3}{2}} = 2.98098$ cubic feet.

When the weir occupies the full width of stream—that is—having no end contractions the formula becomes

$$Q = 3.33\,L\,H^{\frac{3}{2}}$$

It is only in those instances requiring extreme accuracy of discharge that the formula for correction of head due to velocity of approach, need be applied.

Friction of Air in Long Pipes.

The following formulæ supposes the pipe reasonably straight, with no material deviations in direction, and of uniform diameter from end to end. Of course the formulæ may be applied to a combination pipe containing constantly reducing or enlarging divisions, by calculating the head for each division and adding the several heads together for a total friction head.

Let L = length of pipe in feet.

C = discharge in cubic feet per minute.

D = diameter of pipe in inches.

H = head in pounds pressure per square inch necessary to overcome friction alone.

Then—

$$H = \frac{C^2\,L}{(3.7\,D)^5\,83.1} \qquad C = \sqrt{\frac{H\,(3.7\,D)^5 \times 83.1}{L}}$$

$$D = \frac{\sqrt[5]{\dfrac{C^2\,L}{83.1\,H}}}{3.7} \qquad L = \frac{H\,(3.7\,D)^5 \times 83.1}{C^2}$$

It is required to deliver sufficient air at 60 pounds pressure through 3 miles of pipe to develop 3,600 indicated horse power. What pressure will be required at inlet end of a pipe 24 inches diameter.

Let $L = 3$ miles $= 15,840$ feet.

$C = 3,600 \times 33,000$ foot pounds per minute, $144 \times 60 =$ moment per unit of area and travel, of engine piston, and $\frac{118,800,000}{8,640}$ $= 13,750$ cubic feet per minute.

Then—

$$H. = \frac{13,750^2 \times 15,840}{(3.7 \times 24)^5 \times 83.1} = 6.5236 \text{ pounds per square inch.}$$

$$C = \sqrt{\frac{6.5236 \times (3.7 \times 24)^5 \times 83.1}{15,840}} = 13,750 \text{ cubic feet.}$$

$$D = \frac{\sqrt[5]{\frac{13,750^2 \times 15,840}{83.1 \times 6.5236}}}{3.7} = 24 \text{ inches.}$$

$$L = \frac{6.5236 \times (3.7 \times 24)^5 \times 83.1}{13,750^2} = 15,840 \text{ feet.}$$

Suppose the pipe in the example is 18 inches diameter, what then will be the friction head?

$$H = \frac{13,750^2 \times 15,840}{(3.7 \times 18)^5 \times 83.1} = 27,491,$$

or a total head at inlet end of 66.5236 pounds for a 24-inch pipe, and a total head 87,491 pounds for an 18-inch pipe.

VELOCITY OF SOUND.

"In air and other gases, the velocity of sound depends on the pressure, density, and absolute temperature," and the rate is expressed by the formula

$$v = 1092\sqrt{\frac{T}{493.2}}$$

When $v =$ velocity in feet per second, and
$T =$ absolute temperature $=$ observed temperature $+ 461.2$

What is the velocity (v) of sound in an atmosphere at 60 Fahr.?

$$T = 60 + 461.2 = 521.2$$

Then—

$$v = 1092\sqrt{\frac{521.2}{493.2}} = 1122.57 \text{ feet}$$

and in an atmosphere at 10 Fahr.,

$$v = 1092\sqrt{\frac{471.2}{493.2}} = 1067.32 \text{ feet.}$$

QUALITY OF STEAM.

Two general methods are employed to determine the quality or heat power of steam. One, which is the simplest and most readily improvised for immediate use, consists of a tight tub—usually an oil barrel found in most manufacturing establishments—sawn into above the bilge, to carry about 25 or 30 gallons of water and a very sensitive small platform scale, upon which the tub is mounted and carefully balanced for tare.

A pipe, usually ¾-inch, is connected at one end with the steam drum of the boiler, or with the main pipe leading from the boiler—(in which case the small steam pipe must have its internal bend opposite to the direction of flow. That is, if the steam flows from right to left, the bend must be to the right. The other end of the steam pipe depends into the tub, and is furnished with a distributer of many lateral jets, which prevent the blow of steam from influencing the action of the scale. A stop cock or straightway valve in the steam pipe regulates the flow of steam into the tub. The operation is as follows: Suppose a certain weight of water, at normal temperature, is weighed into the tub, and the temperature of the water has been carefully noted with an accurate thermometer, and suppose a known weight of steam is then blown into and condensed by the water, and the temperature of contents of tub is again taken, then the range of temperature with constant weights of water and steam and temperature of normal water is roughly an index of the quality of steam condensed.

To illustrate the problem, let T = normal temperature of water, and S, the specific heat of water at temperature T. Let T_1 = temperature of water after steam has been condensed, and S_1 specific heat of water at temperature T_1; then range of heat is $R = T_1\ S_1 - T\ S$.

Let W = weight of water, and w = weight of steam condensed: H = total heat of steam, and L = heat of vaporization, at observed pressure taken from Regnault's table. Then $W\ R$ = heat added to water,

and $\frac{W\ R}{w}$ = heat added to water per pound of steam condensed, and

$$\frac{W\ R}{w} + T_1\ S_1 = \text{total heat per pound of condensation, and—}$$

$$H - \left(\frac{W\ R}{w} + T_1\ S_1\right) \quad \text{or} \quad \left(\frac{W\ R}{w} + T_1\ S_1\right) - H = \text{discrepancy,}$$

or, excess of heat units per pound of steam condensed, indicating an entrainment of water in the steam or a super heat respectively, and percentage of water entrained in the steam—

$$E = \frac{H - \left(\frac{W\,R}{w} + T_1\,S_1\right) 100}{L}$$

and degrees of super heat in steam—

$$H_1 = \frac{\left(\frac{W\,R}{w} + T_1\,S_1\right) - H}{.475}$$

The second method of determing the quality of steam is by means of a small surface condenser, the coil of which is connected with the steam drum, or boiler, or with main steam pipe as before, and the jacket around the coil connected with a cold water supply.

The data with this arrangement consists of the weight of condensing water, W; weight of condensation, w; temperatures of injection, T, and overflow, T_1 and temperature of condensation as it leaves the condensing worm, T_2. The formula is like that previously given for the simple calorimeter, excepting $T_2\,S_2$ is added to $\frac{W\,R}{w}$ for total heat per pound of condensation.

The formula for determining the specific heat of water, adopted from Rankine, is,

$$S = 1 + .000000309\ (T - 39.1)^2$$

T = any temperature reckoned from zero of Fahrenheit's scale.

The following data is from the contract trial of the Worthington pumping engine, at Buffalo, N. Y., July, 1882:

$$W = 200. \quad w = 10.208. \quad T = 77.208. \quad T_1 = 130.625$$

steam pressure = 57.674 above atmosphere, and range of temperature $R = T_1\,S_1 - T\,S$

$$S = 1 + .000000309\ (77.208 - 39.1)^2 = 1.0004487$$

and $T\,S = 77.208 \times 1.0004487 = 77.242$

$$S_1 = 1 + .000000309\ (130.625 - 39.1)^2 = 1.0025884$$

and $T_1\,S_1 = 130.625 \times 1.0025884 = 130.963$
and $R = 130.963 - 77.242 = 53.721$.

Then heat units added to water, W, per pound of steam condensed was—

$$\frac{200 \times 53.721}{10.208} = 1052.537$$

and heat units per pound of steam—

$$1052.537 + 130.963 = 1183.5$$

The total heat of steam at observed pressure according to Regnault, $H = 1206.85$ and heat of vaporization $L = 899.53$. From which the efficiency of the steam is deduced as—

$$E' = \frac{1183.5}{1206.85} = .98066$$

and percentage of water entrained in the steam—

$$E = \frac{1206.85 - 1183.5}{899.53} \times 100 = 2.5958$$

In making calorimeter tests for quality of steam, great care must be observed in taking weights and temperatures to obtain reliable results.

DIMENSIONS OF STEAM PORTS.

The area of a steam port should be such that the maximum flow will not exceed 100 feet per second. Thus, an eighteen inch cylinder, having an area of 254.47 square inches at 600 feet piston speed, would represent a consumption of $\frac{254.47}{144} \times 10 = 17.6715$ cu. ft. of steam per second, or in this proportion for any point of cut off. The steam port for this engine should be $\frac{17.6715}{100} \times 144 = 25.447$.

According to the following table taken from Auchincloss' Link and Valve Motion, the area of above steam port would be 25.447 square inches. At lower piston speeds the co-efficient produces relatively larger port areas. Thus, for above cylinder and piston speed of 300 feet, the port area, by calculation upon a velocity of flow of 100 feet per second, would be 12.723 square inches, while the co-efficient of table gives an area of 14 square inches. The co-efficients in the table. however, recognize the fact that the perimeter, or frictional surface of a steam port, is inversely as the area, and undertake to provide for this by assuming lower rates of flow per second for the lower piston speed.

Knowing the conditions, however, under which an engine will work a port opening based upon a velocity of flow of 100 feet per second will be ample.

Speed of Piston	*Port Areas.*	*Steam Pipe Area.*
200 feet per min......	.04 area of piston......	.025 area of piston.
250 " "	.047 " "	.032 " "
300 " "	.055 " "	.039 " "
350 " "	.062 " "	.046 " "
400 " "	.07 " "	.053 " "
450 " "	.077 " "	.06 " "
500 " "	.085 " "	.067 " "
550 " "	.092 " "	.074 " "
600 " "	.100 " "	.08 " "

The above co-efficients divided by .75 will give areas of exhaust ports.

H. P. BASED ON INDICATOR DIAGRAMS.

In estimating the power of steam engines from indicator diagrams care should be had in calculating the power of forward and return strokes separately. Thus, an 18 inch piston with a 3 inch rod would present an effective area for forward stroke of 254.47 square inches, and for return stroke of 247.4 square inches. If the mean effective pressure for forward and return stroke are alike (which is seldom or never the case), then the areas may be merged into a mean area and referred to whole piston speed per minute. If they are different, which is the author's experience of many trials of steam engines, then the work of opposite ends of cylinder should be independently computed and added together for indicated power of engine.

To illustrate, suppose for above areas a piston speed of 500 feet, and a mean effective pressure for forward stroke of 28 pounds, and for return stroke 25 pounds, the power due forward stroke

$$\frac{254.47 \times 28 \times 250}{33{,}000} = 53.98 \text{ H. P.}$$

and for return stroke—

$$\frac{247.4 \times 25 \times 250}{33{,}000} = 46.856 \text{ H. P}$$

and indicated horse power of engine 100.836.

Let us reverse the pressures and estimate the power; then for forward stroke we have—

$$\frac{254.47 \times 25 \times 250}{33{,}000} = 48.195 \text{ H. P.}$$

and for return stroke—

$$\frac{247.4 \times 28 \times 250}{33{,}000} = 52.479 \text{ H. P.}$$

and indicated power of engine 100.674.

Averaging pressures and areas in the usual way the result is—

$$\frac{250.935 \times 26.5 \times 500}{33{,}000} = 100.754 \text{ H. P.}$$

Although the differences are not great, for precision the method proposed should be used.

PRESSURE OF VAPOR OF WATER.

Let p = absolute pressure per square inch.

A = constant = 8.2591

B = constant = 2731.618 = Log. 3.43642

C = constant = 396944.7 = Log. 5.59873

T = absolute temperature of water = observed temperature on Fahr. scale + 461.2

Then, by formula adopted from Rankine,

$$\text{Log. } p = A - \left(\frac{B}{T} + \frac{C}{T^2} + \text{Log. } 144\right)$$

Suppose in a digester for the decomposition of fats into fat acids and glycerine, the emulsion (fat and water) is maintained at a constant temperature of 440 Fahr., what is the pressure of vapor corresponding to this temperature?

$$T = 440 + 461.2 = 901.2 = \text{log. } 2.9548212$$

$$\frac{B}{T} = \frac{2731.618}{901.2} = 3.031098$$

$$\frac{C}{T^2} = \frac{396944.7}{901.2^2} = .4887536$$

$$\text{Log. } 144 = 2.1583625$$

Then—

$$\text{Log. } p = 8.2591 - (3.031098 + .4887536 + 2.1583625) =$$

$$\text{Log. } 2.5808859 = p = 380.96 \text{ pounds.}$$

Suppose the temperature 66° Fahr., then—

$$T = 660 + 461.2 = 1121.2 = \text{Log. } 3.0496831$$

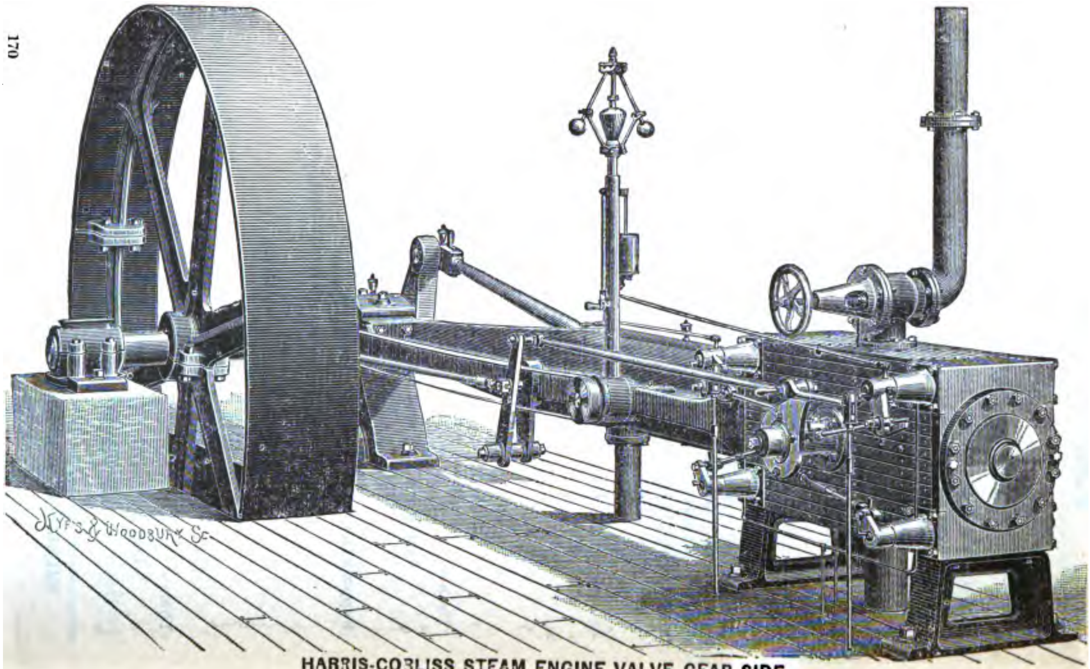
$$\frac{B}{T} = \frac{2731.618}{1121.2} = 2.436408$$

$$\frac{C}{T^2} = \frac{396944.7}{1121.2^2} = .3157621$$

Then—

$$\text{Log. } p = 8.2591 - (2.436408 + .3157621 + 2.1583625) =$$

$$\text{Log. } 3.3485674 = p = 2231.362 \text{ pounds.}$$



HARRIS-CORLISS STEAM ENGINE VALVE GEAR SIDE.

TRIALS OF AUTOMATIC CUT-OFF ENGINES.

It is worthy of note that in all competitive trials of automatic engines where the conditions of performance have been alike for all competitors, the Harris-Corliss has always given the highest economy.

At the fair of the American Institute, New York, October, 1869, the Babcock & Wilcox and Harris-Corliss engines were entered for the trials. Mr. Chas. E. Emery, M. E., conducted the experiments.

The Babcock & Wilcox cylinder was steam jacketed, and the cut-off effected by steam pressure, a small piston in an auxiliary cylinder on the back of the distribution (main) valve, being connected to the cut-off plates, and the regulating mechanism being connected to the small slide valve admitting steam to this cylinder.

The Harris-Corliss cylinder was unjacketed, but covered with non-conducting cement and lagged with wooden staves. The steam valves were operated by the well known Corliss liberating gear and Watt regulator.

The following data is from Mr. Emery's official report:

	Babcock & Wilcox.	Harris-Corliss.
Duration of experiment, hours	8	8
Cylinder, inches	16 × 42	16.13 × 42
Revolutions	60.331	60.277
Pressure in the pipe	81.69	80.51
Cut-off in parts of stroke	.189	.226
Mean effective pressure	31.057	29.728
Indicated horse power	78.792	76.579
Friction horse power, total	10.088	7.480
Net horse power	68.704	68.099
Water per net horse power, per hour	29.231	28.880
Coal per net horse power (estimated), per hr.	3.248	3.209
Coal per ind. horse power (actual) per hour	3.966	3.195
Relative efficiency by steam	0.988	1.000
Relative efficiency by coal	0.805	1.000

At the Cincinnati Industrial Exposition of 1874 the Harris-Corliss and Babcock & Wilcox engines were entered for the trials. The author conducted the experiments.

The Harris-Corliss engine was similar to the one tested at New York. The Babcock & Wilcox engine differed slightly in the manner of working the jacket.

The following data is from the author's report to the Exposition commissioners:

	Harris-Corliss.	Babcock & Wilcox.
Duration of experiment, hours	8	8
Cylinder, inches	16 06 × 48	16 × 30
Revolutions	60.108	84.308
Piston speed	480.86	421.54
Pressure in the pipe	70.477	70.326
Cut-off in parts of stroke	.206	.260
Mean effective pressure	25 45	29.18
Indicated horse power	74 934	74 942
Friction horse power	9.044	13 098
Net horse power	65 890	61 844
Water per net horse power, per hour	43 84	36 64
Coal per net horse power, per hour	3 65	4.07
Relative efficiency	1.000	0.897

At the Cincinnati Industrial Exposition of 1875, the Harris-Corliss and Buckeye Automatic Cut-off engines were entered for the trials. The author, Isaac V. Holmes and J. F. Flagg, as a Board of Experts, conducted the experiments.

The Harris-Corliss was the same engine tested the previous year. The Buckeye engine involved certain novel principles of construction. The distribution and cut-off valve were of the Meyer-Gonzenbach variety; the angular advance of the cut-off eccentric, which was loose on the shaft, being controlled and adjusted in position by a centrifugal regulator keyed to the shaft.

The following data is from the author's report to the commissioners of the Exposition:

	Harris-Corliss.	Buckeye.
Duration of trial, hours	8	8
Cylinder, inches	16 06 × 48	12 × 20
Revolutions	58.537	186.111
Piston speed	468.30	453.70
Pressure in the pipe	74 38	75.79
Cut-off in parts of stroke	.175	.151
Mean effective pressure	25.153	28 923
Indicated horse power	72 328	37.198
Friction horse power	9.186	4.167
Net horse power	63.142	33.031
Water per net horse power, per hour	26.498	28.153
Coal per net horse power, per hour	2.944	3 128
Relative efficiency	1.000	0.941

During August, 1877, the author conducted a series of economy trials on a Harris-Corliss automatic engine for the Messrs. Gibson, flour millers, Indianapolis. The object of the trials was to determine the gain in economy due to operating the engine condensing. The engine was furnished to run non-condensing, and the condenser and air pump were added a few weeks prior to the tests.

The following data is from the author's report to the proprietors of the mill:

	Condensing.	Non-condensing.
Duration of trial, hours	8	8
Cylinder, inches	18 × 42	18 × 42
Revolutions	74.288	73.600
Piston speed, feet	520.02	515.20
Pressure in the pipe, pounds	58.50	76.37
Vacuum by gauge, inches	21.83	......
Cut-off in parts of stroke	.108	.189
Mean effective pressure, pounds	26.928	29.471
Indicated horse power*	105.47	115.43
Friction horse power, total	12.64	13.07
Net horse power	92.83	102.36
Water per indicated horse power, per hr., pds.	18.593	25.391
Coal per indicated horse power, per hr., pds.	2.066	2.821
Relative efficiency	1.000	0.733

* The greatest amount of work was during non-condensing run. The amount of grain elevated and barrels of flour manufactured for each run was nearly in the ratio of the net power.

DESCRIPTION OF THE TRIAL FOR ECONOMY OF A HARRIS-CORLISS CONDENSING ENGINE IN A FLOURING MILL.

[A. A. Freeman & Co., at LaCrosse, Wis., taken from author's report to the proprietors, A. A. Freeman & Co., New York.]

The engine, 24″ diameter of cylinder and 60″ stroke of piston, is condensing, and fitted with the ordinary jet condenser and reciprocating air pump. The injection water is obtained by a lift of 15′ from the Mississippi river, upon the bank of which the mill stands; and during the trial the condensing water entered the injection pipe, at a temperature near the freezing point. The steam valves were formerly closed by the usual weights, but previous to the trial vacuum dash pots were added to insure a prompt closing of the valve when liberated from the hook. The engine is furnished with a pulley fly-wheel 20′ diameter and 32″ face; driving back to the line shaft with a 30″ double leather belt.

The exhaust of engine is closely connected to the condenser by a 10″ pipe, and steam is conveyed from the boiler by a 7″ pipe.

Steam is furnished by a pair of tubular boilers set in battery, and each of the following dimensions: 60″ diameter of shell, 12′ long, 50-4″ tubes. Each boiler is fitted with a vertical steam dome, 30″ diameter x 36″ high, and over these and joined to them by short legs is a horizontal steam drum, 24″ diameter and 14′ long.

The steam pipe is joined by branch pipes to the side of the horizontal drum.

The feed water is taken from a drop leg in the overflow pipe from the condenser, and conducted to the suction of a single acting plunger pump driven from the engine by belt. Into the breeching or front smoke connection has been introduced a fuel economizer, consisting

of 250′ of 2½″ iron pipe, through which the feed water is forced to the boiler.

The furnace is arranged to burn slabs and hard wood, although by the record it would appear to be well adapted for coal (the fuel used during the trial of engine). The lack of a suitable bridge wall, and the very large furnace doors and grate surface are not calculated for maximum economy with coal as a fuel; and it is eminently probable that with a different construction of furnace the efficiency of the boilers during the trial of engine would have been higher.

The entire net power of engine is expended in driving the machinery of the mill, which consists of twelve run of 54″ buhrs, and three run of 48″ buhrs; two crushing rolls, each with 3—12″x30″ cylinders; five rolls, each with 2—12″x30″ cylinders, and one roll with 2—12″x18″ cylinders. The bolting machinery consists of one chest with two reels; two chests with three reels; one chest with six reels, and one chest with eight reels; in all twenty-two bolting reels and forty-eight conveyors.

The cleaning machinery consists of two "cockle" machines; one "scouring" machine; one "separator," and two brushing machines. Of the purifying machines there are seventeen, and one shaking machine; four flour packers; four stand of wheat elevators; four stand of flour elevators, and twenty-one middlings elevators. One small and two large exhaust fans.

To this should be added the machinery of the grain elevator, which is driven by belt from the third story of the mill; and the line shafting, connecting belts, pulleys, and gearing, forming the general machinery of the mill.

In the following tables are given the principal measured and calculated data of engine and boilers. The clearance was not measured, but estimated at three per cent. of piston displacement, this being the usual clearance in Harris-Corliss engines of like dimensions.

The factor of horse-power due mean area, and velocity of piston for each mean effective pound pressure has been calculated as follows: The area of a 24″ piston is 452.39, sq. ins. and the area of the rod (3.375″) is 8.9462 sq. ins., and the mean area of piston is, therefore,

$$452.39 - \frac{8.9462}{2} = 447.917 \text{ sq. ins., and the factor of horse power.}$$

$$\frac{447.917 \times 596.166}{33{,}000} = 8.20446$$

The valve functions have been measured on the diagrams. The volume of steam accounted for to release is obtained by taking the mean area (feet) of piston into the piston travel (feet) per hour to point of release, to which is added the hourly volume of clearance.

The volume of steam retained by exhaust closure is obtained by taking the mean area of piston, in feet, into the travel of piston, in feet, per hour, from exhaust closure to end of stroke, to which is added the hourly volume of clearance.

The dimensions of boilers and fire grates are furnished by your engineer (Storey), from which have been deduced the heating surface, grate surface and calorimeter of tubes, and ratios of heating to grate surface, and grate surface to cross section of tubes.

DIMENSIONS OF ENGINE.

Cylinder	Unjacketed.
Diameter of cylinder	24 inches.
Stroke of piston	60 "
Revolutions per minute during trial	59.616
Piston speed " " " "	596.166 feet.
Factor of H. P. due area and velocity of piston	8.204
Piston stroke to release in parts of stroke	99 370
" " to exhaust closure in parts of stroke	6 067
Clearance (estimated) in parts of stroke	3.000
Volume of steam to release per hour	115038 04 cu. ft.
" " retained by cushion per hour	10189 02 cu. ft.
Diameter of air pump	12 inches.
Stroke of air "	15 "
Diameter of driving pulley	20 feet
Face " " "	32 inches.
Weight " " "	40,000 pounds.

DIMENSIONS OF BOILERS.

Number	2	
Diameter of shells		60 inches.
Length " "		12 feet.
Tubes, each boiler		50-4 inches.
Heating surface shells (2)	250 56	
" " tubes (100)	1245 64	
" " heads (4)	40.72	
" " total		1536.92 sup. ft.
Grate "		51.75 sup. ft.
Calorimeter of flues		1256.64 sup. in.
Heating to grate surface		29.70
Grate surface to calorimeter		5.93

The trial of engine for economy of performance and trial of boilers for evaporative efficiency were made simultaneously (March 18); all preparations having been completed, the trial began at 9:15 A. M., and terminated at 7:15 P. M.; duration of trial, ten hours.

The test of boiler efficiency was with coal.

The load was that usually carried in the daily operation of the mill, and through the care of your chief miller (Lang) was held quite uniform during the ten hours run. It is possible that the mean power developed is slightly greater than usual, from the fact that the operatives were cautioned to avoid breaks in the load, and that they obeyed

the injunction is best attested by the indicator diagrams, which exhibit but slight variations in the power during the economy trial.

The diagrams were taken by independent indicators, one to each end of cylinder. Forty (40) springs were used, and the drums were moved by well constructed bell cranks, and reciprocating connections hung on a stout gallows frame. The joints of the levers and connections were carefully made, and means were provided to take up wear, and avoid lost motion.

The strings on the indicator barrels were only long enough to couple with the pins on the short stroke reciprocating bar, and the recoil springs were adjusted as nearly as possible to the same tension. The length of diagrams was uniformly 4.78″.

During the trial a pair of diagrams were taken regularly every fifteen minutes, making eighty-two diagrams from which has been obtained the initial pressure in cylinder; piston stroke to cut off; ratios of expansion by pressures and by volumes; terminal pressure; counter pressure at mid-stroke; utilization of vacuum, and mean effective pressure on the piston, from which is obtained the mean power developed.

The vacuum in the condenser and the pressure in the boilers were taken from gauges in the engine-room regularly every fifteen minutes.

The temperature of water to the condenser was taken in the river at the mouth of the injection pipe. The temperature of overflow from the condenser was taken in the measuring tank. The temperature of feed to the boiler was taken in the feed pipe near the check valves.

The water delivered to the boilers was measured in the following manner: Two oil barrels were carefully washed inside and placed on the same level in the engine-room; to the bottoms of these was connected by branch pipes, the suction pipe of pump; each branch being provided with an open way cock to shut off the flow when the level had been reduced to the lowest gauge point. The pipe from the hot well to the pump was cut and carried out over the barrels; a connection made by branches to each barrel, and a stop valve in each branch regulated the flow of water into the tanks. The tanks or barrels were numbered one and two, and were alternately filled to the overflow notch in the rim, and emptied to the center of the branch pipe in the side of barrel, and the contents discharged into the pipe leading to the pump.

Whilst the number one barrel was running out, the number two barrel was filling with water from the hot well, and directly the first barrel was emptied to the lower gauge point, it was turned off; and the second barrel turned on; and so on during the entire trial; the

empty barrel being shut off before the full one was turned on, to prevent transfer of water from the full to the empty barrel. Directly each barrel of water was turned on, the time was entered in the log, and a tally made by the assistant in charge of the tanks. From time to time my record of tanks discharged was compared with the assistant's tally to avoid error in the count.

After the trial, the capacity of each tank was determined by filling to the overflow notch, noting temperature, drawing off to the lower gauge point and weighing. The temperatures of the tanks of water discharged into the suction pipe of feed pump, having been regularly noted during the trial; the weight of water delivered to the boiler was deduced from the number of tanks discharged, into the weight of tanks at mean observed temperature.

The calorimeter tests of water entrained were made by drawing off from the steam drum, near the pipe to the engine, a given weight of evaporation, and condensing it in a given weight of water, noticing the temperature of the water before and after the steam was turned in, and the pressure of evaporation each time an observation was made. The thermal values due the ranges of temperature and the weights of steam and water, together with the thermal values of saturated steam at observed pressures, constituted the data from which has been estimated the heat units resident in a pound of evaporation during the trial, from which has been deduced the water entrained in the steam as 12.84 per cent. of the total water pumped into boilers. Twenty calorimeter observations were made during the ten hours' trial.

The revolutions of the engine are nominally 60 per minute; but from the ten hours' continuous record by counter, the mean revolutions per minute was 59.616.

The coal fired during trial of engine was Wilmington, mined in the northern part of Illinois, and from the evaporative efficiency developed, of very fair quality.

The ash pit and fires were cleaned before trial, and the ash and clinker accumulated during the ten hours' firing weighed back dry. The non-conbustible by weight constituted 7.3 per cent. of the total coal fired. Previous to commencement of run, the water level in both boilers was marked on the glass gauges, and the fires leveled and thickness noted: the same conditions of fires and water level obtained at the end of trial.

In the following tables are given the observed and calculated data, illustrating the performance of engine and boilers. All data from the diagrams are means of eighty-two readings, and all other data are means of forty-one readings.

The economy of engine by steam and by coal is developed upon the mean quantities charged per hour.

DATA FROM TRIAL OF ENGINE.

Date of trial	March 13, 1879.
Duration of trial	10 hours.
Mean pressure by boiler gauge above atm	92.876 lbs.
" initial pressure above atm	89.376 lbs.
" terminal " absolute	12.018 lbs.
" counter " "	2.696 lbs.
" cut off in parts of stroke apparent	15.560
" " " " actual	18.019
" vacuum by gauge	26.40 inches.
" " " diagrams	24.05 "
" temperature of injection	33.840
" " of hot well	92.725
" effective pressure	32.9792 lbs.
Indicated horse-power	270.5796
Ratio of expansion by volumes	5.549
" " " " pressures	8.643

"ECONOMY OF ENGINE."

Total water per hour to boilers	5037.128 lbs.
Water (steam) per hour to calorimeter	10.000 lbs.
" entrained per hour in the steam	655.583 lbs.
Net steam per hour to engine	4371.545 lbs.
Steam per indicated horse-power, actual	16.156 lbs.
" " " " " by the diagrams	13.035 lbs.
Per centage of steam accounted for	80.682
Coal burned per hour	535. lbs.
Coal per indicated horse-power per hour	1.9772 lbs.
" " " " evaporation 9 to 1	1.7950 lbs.
Combustible, per indicated horse-power, per hour	1.8328 lbs.

PERFORMANCE OF BOILERS.

Date of trial	March 13, 1879.
Duration of trial	10 hours.
Pressure by gauge	92.876 lbs.
Temperature of feed to heater	92.725
" " " " boiler	114.324
Elevation of feed by heater	21.599
Percentage of gain by heater	1.723
Total water pumped into boilers	50371.28 lbs.
" " entrained in the steam (12.84 per cent)	6467.70 lbs.
" steam furnished	43903.58 lbs.
" coal fired	5350. lbs.
" non-combustible weighed back	390. lbs.
" combustible	4960. lbs.
Steam per pound of coal	8.206 lbs.
" " " " combustible	8.852 lbs.
" " " " coal from temp. of 212 and pres. of atm	9.639 lbs.
Steam per square foot of heating surface per hour	3.022 lbs.
Coal " " " " grate " " "	10.300 lbs.
Percentage of ash in coal	7.3
Coal burned during trial	Wilmington, Illinois.

During the economy trial of engine, the flour manufactured was, by the miller's report, 217 barrels high grade, and 2 per cent. added for low grade, or 221.34 barrels produced in ten hours. The mean indicated power of engine was 270.56 horse-power, and the hourly expend-

iture of power per barrel of flour produced was $\frac{270.56}{22.134} \doteq 12.224$ H. P.

The coal burned for whole trial was 5350 pounds, and coal per barrel of flour produced becomes $\frac{5350}{221.34} = 24.198$ pounds.

Whilst the experiments of firing slabs and hard wood were in progress, the engine was indicated for distribution of the power in the mill.

The first (*A*) load was with all the machinery on, and operating under the ordinary conditions. The second (*B*) load was with all the machinery on, except the machinery in elevator building. The third (*C*) load was with all the machinery on, except the flour packers. The fourth (*D*) load was with all the machinery on, except the cleaning machinery and flour packers. The fifth (*E*) load was with all the machinery on, except the crushing rolls. The sixth (*F*) load was with all the machinery on, except the purifiers, and the seventh (*G*) load was with all the machinery on, except the grinding buhrs.

The changes of load were made quickly in order to preserve the conditions of ordinary performance in the special machinery driven; and the power developed for each load has been estimated from six diagrams, three from each end of cylinder.

The indicated loads were as follows:

First load A	267.503	H. P.
Second load B	262 585	"
Third load C	263 706	"
Fourth load D	250.726	"
Fifth load E	246.740	"
Sixth load F	243.645	"
Seventh load G	117.149	"

Each of these loads is made up of the friction of engine in all parts extra friction of engine due to the load; friction of all the driving machinery in the mill, and power required to drive the special machinery, including friction; in like manner the differences between the maximum load and reduced loads nearly represent the power required to drive the special machinery not on, including its own friction.

The extra friction of the engine is a certain co-efficient of the load actually carried, and, of course, in quantity varies with the load; hence the difference between the maximum load and lesser loads represents slightly more than the power actually absorbed by the special machinery not driven.

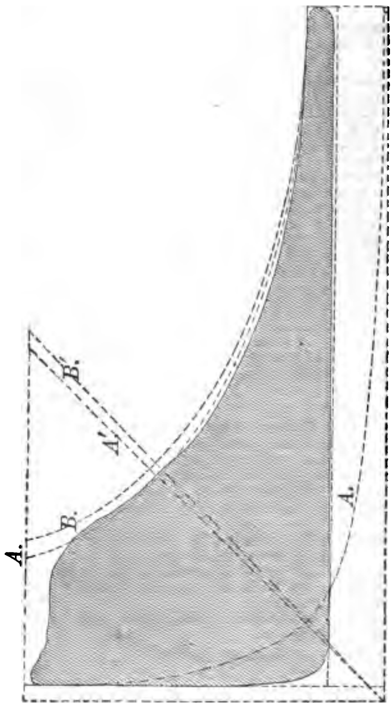
From the several independent loads I deduce the distribution of the power in the mill as follows:

Total indicated power of engine load (*A*)		267.503
Friction of engine alone	16.409	
Extra friction due load	12.554	
Grinding buhrs	150 354	
Cleaning machinery	12 980	
Elevator	4.918	
Crushing rolls	20.763	
Bolting reels, conveyors, fans and general machinery	21.860	
Middlings purifiers	23.868	
Flour packers	3.797	
		267.503

INDICATOR DIAGRAM

FROM 20″ × 48″ HARRIS-CORLISS ENGINE,

FLOUR MILLS OF W. TROW & CO., MADISON, IND.



Data.

$A\ A$ = isothermal curves $= p \propto v$

B = adiabatic curve $= p \propto v^{\frac{10}{9}}$

A' = axis isothermal curves.

B' = axis adiabatic curve.

THE HARRIS-CORLISS ENGINE,

at the Millers' International Exhibition, Cincinnati, June, 1880.

Three engines, the Harris-Corliss, Reynolds-Corliss, and Wheelock, were submitted to test trials, of which the former developed the best average economy, condensing and non-condensing.

The following data relating to the performance of the Harris-Corliss engine is taken from the author's reports to the Commissioners of the Exibition.

	HARRIS-CORLISS CONDENSING.	HARRIS-CORLISS NON-CONDENSING.
Date of trial	June 21.	June 22.
Duration of trial, hours	10	10

GENERAL OBSERVATIONS.

Steam pressure at engine	91.65	91.48
Barometer	29.55	29.55
Vacuum by gauge	25.67	
Temperature of air	87.60	85.30
" " injection	75.90	
" " hot well	97.50	
Revolutions per minute	75.83	75.81

FROM THE INDICATOR DIAGRAMS.

Initial pressure	90.072	89.522
Cut-off in decimal of stroke	0.11867	0.13627
Pressure at cut-off	86.966	85.910
Terminal pressure, absolute	14.568 (absolute)	17.037 (above atm.)
Counter pressure at mid stroke	3.352	0.415
Vacuum at mid stroke	11.152	
Maximum compression pressure	26.595	46.098
Mean effective pressure	35.6722	28.9397

RATIO OF EXPANSION.

By volumes	6.9505	6.3496
" pressures	6.9403	6.1059
Theoretical cut-off	0.14386	0.15748
Relative gain by expansion	2.33	1.55

LOAD.

Indicated horse power	165.5781	134.2926

DISTRIBUTION OF LOAD.

Friction of engine	9.5734	9.5609
Gross load	156.0047	124.1317
Extra friction of engine due load	6.2402	4.9893
Power absorbed by air pump	4.6879	
Net effective horse power	145.0766	119.27
Co-efficient of useful effect	0.8762	0.8916

STEAM EXPENDED.

Water weighed to boilers, pounds	32,296.	32,708
Leakage of weighing tanks	13.	
Correction for variation of water level.	+ 285 69	
Condensed in calorimeter	505 50	547.75
Net steam delivered to engine	32,063.19	32,160.25

ECONOMY OF ENGINE.

Steam per indicated horse power per hour corrected for relative value of steam	19.3642	22.0541
Coal per indicated horse power per hour, evaporation 10 to 1	1.9364	2.2054
Steam per hour by the diagrams	2277.581	2419.078
Percentage of steam accounted for	71.034	75.346
Steam per indicated horse power per hour by the diagrams	13.755	18.013

CONDENSING WATER.

Water expended per pound of steam condensed, gallons	3.9

In the many public competitive trials of steam engines the Harris-Corliss has always led all competitors, and of the many well conceived attempts to produce an engine which would achieve a higher economy, not one, up to the present time, has realized the hopes of its projectors.

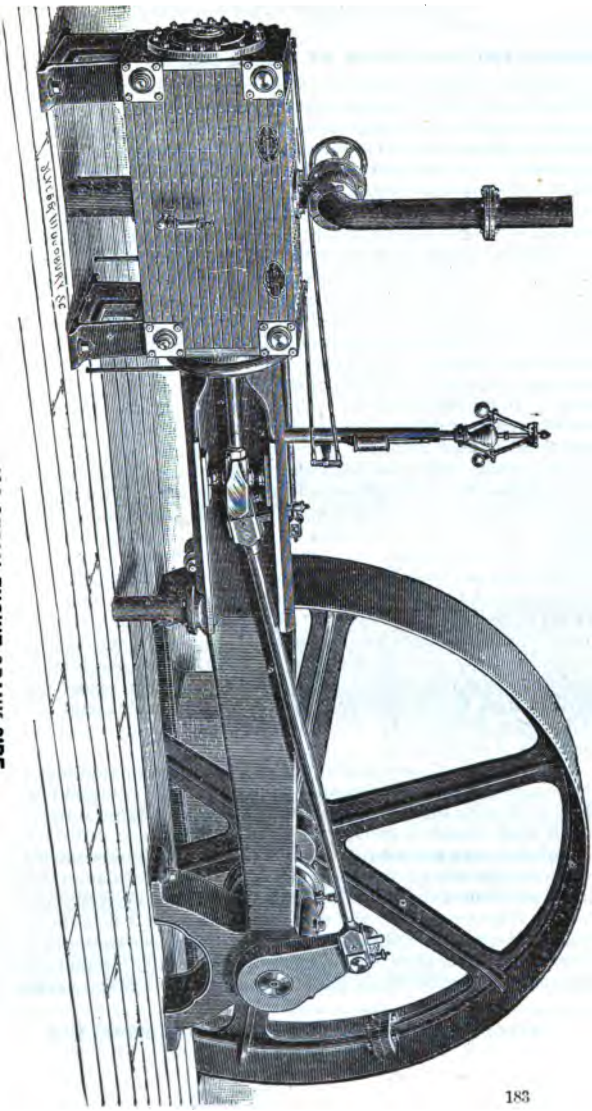
No comparison can be instituted between the Harris-Corliss and other automatic engines; none approach it in excellence near enough to justify comparison. While other engines have yielded good results, the Harris-Corliss has given better.

No engine with single cylinder, unjacketed, has given the result in point of economy shown in the test trial of Harris-Corliss engine at La Crosse. (*See page* 173.)

In the competitive trials at the Fair of the American Institute, 1869, the Harris-Corliss engine beat the Babcock & Wilcox by 35 per cent.

In the competitive trials at the Cincinnati Industrial Exposition, 1874, the Harris-Corliss engine beat the Babcock & Wilcox 11.5 per cent.

In the competitive trials at the Cincinnati Industrial Exposition, 1875, the Harris-Corliss engine beat the Buckeye 8 per cent.



HARRIS-CORLISS STEAM ENGINE CRANK SIDE.

REGULATING MECHANISM OF HARRIS-CORLISS STEAM ENGINE.

The great success of the Harris-Corliss engine lies chiefly in the simplicity and precise action of the governing elements; the governor is an independent mechanism saddled with no extraneous load, and free to instantly respond to variations in the angular velocity of rotating parts. (The slightest variation in the angular motion of the shaft or fly-wheel is immediately appreciated by the governor, and a corresponding point of cut-off is instantly indicated.) "An automatic cut-off engine is one in which the volume of steam cut off in the cylinder is exactly proportioned to the steam pressure and imposed load, to automatically regulate the speed of the engine. If the load is increased the piston stroke to cut off is lengthened; if the steam pressure is increased, the piston stroke to cut off is shortened and *vice versa*, and the regulation of cut-off for any stroke depends upon the conditions existing during that stroke. Thus each stroke of the piston and each semi-revolution of the crank possesses a perfect autonomy." In the Harris-Corliss engine, when the steam port is opened for admission of steam to the cylinder, no obstruction exists to the free flow of steam from the boiler, and when the connecting pipe is of proper size. with few bends and well protected from loss of heat by radiation, the initial pressure in the cylinder is within a pound or two of the pressure in the boiler. When steam flows into the cylinder the piston advances with a velocity proportional to the load on the engine and steam pressure. the motion of the piston is communicated to the crank and from the shaft to the governor, and a point of cut-off is indicated for that stroke, the nearness of the steam and exhaust valves to the bore of the cylinder, the prompt opening and instantaneous closing of steam valves, the rapid opening of exhaust and the tightness of valves under pressure, all contribute to the remarkable performance of this engine. The motion of steam and exhaust valves derived from the wrist-plate is peculiar to this engine and next to the precise action of the regulator, has much to do with the high economy of performance.

In other types of automatic cut-off engines, the regulator, instead of the simple duty of governing as to point of cut-off, is obliged to move the cut-off valve through varying spaces against varying resistances, and if made powerful enough to do the latter without disturbing its equilibrium as a governor, the inertia of the governing elements becomes so great as to prevent its proper action for the regulation of speed and graduation of cut-off, and an *uneconomical use of steam consequently follows*. It has been urged, and with apparent reason, that an automatic cut-off governor saddled with *actuation*, as well as *indication* of cut-off, is desirable rather than otherwise, as the graduating elements of the governor are constantly in vibration and respond more quickly to

variations in velocity of rotation. This is true, but when we consider that *actuation* in these cases means moving a heavy cut-off valve against widely varying moments of friction, the advantages of *actuation* combined with *indication* of cut-off disappears. In the Harris-Corliss engine the sole duty of the governor is to indicate the point of cut-off, and actuation is performed by other and independent mechanism; the friction of the governor is inappreciably small and practically constant; gravity furnishes the centripetal force, and the graduating elements are constantly in motion relative to their axes of oscillation, and the regulator *quickly responds to the slightest variation in velocity of rotating parts.*

TABLE OF MEAN EFFECTIVE PRESSURE.

For Different Initial Pressures and Cut-offs.

	CUT-OFF IN PARTS OF PISTON STROKE.*					
Initial Pressure †	.10	.15	.20	.25	.30	.35
50 = 65 absolute	9 105	15.213	20.403	24.881	28 782	32.191
60 = 75 " ...	12.891	19 938	25.926	31.094	35 594	39.528
70 = 85 "	16 676	24 663	31 450	37.307	42 407	46.866
80 = 95 "	20 461	29.389	36.973	43.520	49.220	54.202
90 = 105 "	24 247	34 113	42 497	49.733	56.033	61.540
100 = 115 "	28.032	38.838	48.019	55 946	62.846	68.878

This table has been calculated for the Harris-Corliss engine, and will be approximately correct only for such other automatic engines as present precisely similar conditions of performance. The clearance has been taken at .025 piston development, and the total stroke at 1.025. While the cut-offs given at the head of the table are the apparent cut-offs, they are in fact as follows: .125— 175—.225—.275—.325—.375. It is assumed that the loss of mean effective pressure by cushion, is compensated by the re-evaporation during latter part of stroke, in an unjacketed cylinder: and that the initial pressure remains constant during admission; then let H represent the hyperbolic logarithm of the ratio of expansion, + 1., P the initial pressure, and h the ratio of expansion, then $\frac{H\ P}{h}$ = mean effective pressure, from which subtract 15 for pressure of atmosphere, and .5 pound for mean counter-pressure.

* Engine worked non-condensing; if engine is worked condensing add 13 75 pounds to the value by the table; thus 70 pounds, cut-off at .20 engine condensing, 31 450 + 13 75 = 45 20 pounds.

† Pressure in the cylinder during admission.

STEAM TABLE.

Pressure by gauge.	Tot'l pres. p'ds.	Inches of Merc'y	Temp. Fahr.	Total heat by pound	Latent heat by pound.	Heat in water by pd.	Relative volume	Weight per cu. ft.
	1	2.036	102.	1145.05	1042.96	102.08	17983.	.00347
	2	4.072	126.27	1152.45	1026.01	126.44	10353.	.00602
	3	6.108	141.62	1157.13	1015.25	141.87	7283.8	.00856
	4	8.144	153.07	1162.62	1007.23	153.39	5608.4	.01112
	5	10.180	162.33	1163.45	1000.73	162.72	4565.6	.01366
	6	12.216	170.12	1165.83	995.25	170.57	3851.0	.01619
	7	14.252	176.91	1167.89	990.47	177.42	3330.8	.01837
	8	16.288	182.91	1169.72	986.24	183.48	2935.1	.02125
	9	18.324	188.32	1171.37	982.43	188.94	2624.1	.02377
	10	20.360	193.24	1172.87	978.96	193.92	2373.0	.02628
	11	22.396	197.77	1174.26	975.76	198.49	2166.3	.02880
	12	24.432	201.96	1175.53	972.80	202.74	1993.0	.03130
	13	26.468	205.88	1176.73	970.02	206.71	1845.7	.03380
	14	28.504	209.56	1177.85	967.43	210.43	1718.9	.03629
.304	15	30.540	213.02	1178.91	964.97	213.94	1608.6	.03878
1.304	16	32.576	216.30	1179.91	962.66	217.25	1511.7	.04123
2.304	17	34.612	219.41	1180.86	960.45	220.41	1426.2	.04374
3.304	18	36.648	222.38	1181.76	958.34	223.42	1349.8	.04622
4.304	19	38.684	225.20	1182.63	956.34	226.28	1281.1	.04868
5.304	20	40.720	227.92	1183.45	954.41	229.04	1219.7	.05119
6.304	21	42.756	230.51	1184.25	952.57	231.67	1163.8	.05360
7.304	22	44.792	233.02	1185.01	950.79	234.22	1112.9	.05605
8.304	23	46.828	235.43	1185.74	949.07	236.67	1066.3	.05851
9.304	24	48.864	237.75	1186.45	947.42	239.03	1023.6	.06095
10.304	25	50.900	240.00	1187.14	945.82	241.31	984.23	.06338
11.304	26	52.936	242.17	1187.80	244.28	243.52	947.86	.06582
12.304	27	54.972	244.28	1188.44	942.77	245.67	914.14	.06824
13.304	28	57.008	246.33	1189.07	941.32	247.75	882.80	.07067
14.304	29	59.044	248.31	1189.67	939.90	249.77	853.60	.07308
15.304	30	61.080	250.24	1190.26	938.92	251.74	826.32	.07550
16.304	31	63.116	252.12	1190.83	937.19	253.64	800.79	.07791
17.304	32	65.152	253.95	1191.40	935.88	255.52	766.83	.08031
18.304	33	67.188	255.73	1191.94	934.61	257.33	754.31	.08271
19.304	34	69.224	257.46	1192.47	933.36	259.11	733.09	.08510
20.304	35	71.260	259.17	1192.99	932.15	260.84	713.08	.08749
21.304	36	73.296	260.83	1193.49	930.96	262.53	694.17	.08987
22.304	37	75.331	262.46	1193.99	929.81	264.18	676.27	.09225
23.304	38	77.367	264.04	1194.47	928.67	265.80	659.31	.09462
24.304	39	79.403	265.60	1194.94	927.56	267.38	643.21	.09700
25.304	40	81.439	267.12	1195.41	926.47	268.94	627.91	.09936
26.304	41	83.475	268.61	1195.86	925.40	270.46	613.34	.10172
27.304	42	85.511	270.07	1196.31	924.36	271.95	599.46	.10407
28.304	43	87.547	271.51	1196.75	923.33	273.42	586.23	.10642
29.304	44	89.583	272.91	1197.18	922.32	274.86	573.58	.10877
30.304	45	91.619	274.29	1197.60	921.33	276.27	561.50	.11111
31.304	46	93.655	275.65	1198.01	920.36	277.65	549.94	.11344
32.304	47	95.691	276.99	1198.42	919.40	279.02	538.87	.11577
33.304	48	97.727	278.30	1198.82	918.47	280.35	528.25	.11810
34.304	4	99.763	279.58	1199.21	917.54	281.67	518.07	.12042
35.304	5	101.799	280.85	1199.60	916.63	282.97	508.29	.12273

STEAM TABLE—Continued.

Pressure by gauge.	Tot'l pres. p'ds.	Inches of Merc'y	Temp. Fahr.	Total heat by pound.	Latent heat by pound.	Heat in water by pd.	Relative volume	W'ght per cu. ft.
36.304	51	103.84	282.10	1198.98	915.74	284.24	498.89	.12505
37.304	52	105.87	283.32	1200.35	914.86	285.50	489.85	.12736
38.304	53	107.91	284.53	1200.72	913.99	286.73	481.15	.12966
39.304	54	109.94	285.72	1201.08	913.13	287.95	472.77	.13196
40.304	55	111.98	286.89	1201.44	912.29	289.15	464.69	.13428
41.304	56	114.02	288.05	1201.80	911.46	290.34	456.90	.13652
42.304	57	116.05	289.11	1202.14	910.64	291.50	449.38	.13883
43.304	58	118.09	290.32	1202.49	909.83	292.65	442.12	.14111
44.304	59	120.12	291.42	1202.82	909.03	293.79	435.10	.14338
45.304	60	122.16	292.52	1203.16	908.25	294.91	428.32	.14566
46.304	61	124.19	293.60	1203.49	907.47	296.02	421.75	.14792
47.304	62	126.23	294.66	1203.81	906.70	297.11	415.40	.15018
48.304	63	128.27	295.71	1204.13	905.95	298.18	409.25	.15244
49.304	64	130.30	296.75	1204.45	905.20	299.25	403.29	.15469
50.304	65	132.34	297.78	1204.76	904.46	300.30	397.51	.15694
51.304	66	134.37	298.79	1205.07	903.73	301.34	391.90	.15919
52.304	67	136.41	299.79	1205.38	903.01	302.37	386.47	.16130
53.304	68	138.45	300.77	1205.68	902.30	303.38	381.18	.16366
54.304	69	140.48	301.75	1205.97	901.60	304.37	376.06	.16590
55.304	70	142.52	302.72	1206.27	900.90	305.37	371.07	.16812
56.304	71	144.55	303.67	1206.56	900.21	306.35	366.24	.17035
57.304	72	146.59	304.62	1206.85	897.53	307.32	361.53	.17256
58.304	73	148.63	305.55	1207.13	898.85	308.28	356.95	.17478
59.304	74	150.66	306.47	1207.42	898.19	309.23	352.49	.17690
60.304	75	152.70	307.39	1207.69	857.53	310.16	348.15	.17919
61.304	76	154.73	308.29	1207.97	896.88	311.09	343.93	.18139
62.304	77	156.77	309.18	1208.24	896.23	312.01	339.81	.18359
63.304	78	158.81	310.07	1208.51	895.59	312.92	335.81	.18578
64.304	79	160.84	310.94	1208.78	894.95	313.82	331.89	.18797
65.304	80	162.88	311.81	1209.04	894.33	314.71	328.08	.19015
66.304	81	164.91	312.67	1209.30	893.71	315.59	324.37	.19233
67.304	82	166.95	313.52	1209.56	893.09	816.47	320.74	.19451
68.304	83	168.99	314.36	1209.82	892.49	317.33	317.20	.19668
69.304	84	171.02	315.19	1210.07	891.88	318.19	313.74	.19885
70.304	85	173.06	316.02	1210.33	891.29	319.04	310.36	.20101
71.304	86	175.09	316.84	1210.58	890.69	319.89	307.07	.20317
72.304	87	177.13	317.65	1210.83	890.11	320.72	303.85	.20532
73.304	88	179.17	318.45	1211.07	889.52	321.54	300.70	.20747
74.304	89	181.20	319.25	1211.31	888.95	322.36	297.62	.20962
75.304	90	185.24	320.04	1211.55	888.38	323.17	294.61	.21185
76.394	91	185.27	320.82	1211.79	887.81	323.98	291.66	.21390
77.304	92	187.31	321.58	1212.03	887.25	324.78	288.78	.21603
78.304	93	189.35	322.36	1212.26	886.69	325.57	285.96	.21816
79.304	94	191.38	323.13	1212.49	886.13	326.36	283.21	.22029
80.304	95	193.42	323.88	1212.72	885.59	327.13	280.50	.22241
81.304	96	195.45	324.63	1212.95	885.04	327.91	277.86	.22153
82.304	97	197.49	325.38	1213.18	884.50	328.68	275.27	.22675
83.304	98	199.53	326.11	1213.40	883.97	329.43	272.73	.22873
84.304	99	201.56	326.84	1213.63	883.44	330.19	270.24	.23085
85.304	100	203.60	327.57	1213.85	882.91	330.94	267.80	.23296

STEAM TABLE—Continued.

Pressure by gauge.	Tot'l pres. p'ds.	Inches of Merc'y	Temp. Fahr.	Total heat by pound	Latent heat by pound	Heat in water by pd.	Relative volume	Weight per cu. ft.
86.304	101	205.64	328.29	1214.07	882.39	331.68	265.81	.23503
87.304	102	207.67	329.00	1214.28	881.87	332.41	263.07	.23715
88.304	103	209.71	329.71	1214.50	881.35	333.15	260.77	.23924
89.304	104	211.74	330.42	1214.71	880.85	333.86	258.52	.24132
90.304	105	213.78	331.11	1214.93	880.34	334.59	256.31	.24340
91.304	106	215.82	331.80	1215.14	879.84	335.30	254.14	.24548
92.304	107	217.85	332.49	1215.35	879.34	336.01	252.01	.24756
93.304	108	219.89	333.17	1215.55	878.84	336.71	249.92	.24963
94.304	109	221.92	333.85	1215.76	878.35	337.41	247.87	.25169
95.304	110	223.96	334.52	1215.97	877.86	338.11	245.86	.25375
96.304	111	225.99	335.19	1216.17	877.38	338.79	243.88	.25581
97.304	112	228.03	335.85	1216.38	876.90	339.48	241.94	.25786
98.304	113	230.07	336.51	1216.58	876.42	340.16	240.03	.25991
99.304	114	232.10	337.16	1216.77	875.94	340.83	238.15	.26204
100.304	115	234.14	337.81	1216.97	875.47	341.50	236.31	.26400
101.304	116	236.17	338.46	1217.17	875.00	342.17	234.50	.26611
102.304	117	238.21	339.10	1217.36	874.54	342.83	232.70	.26816
103.304	118	240.25	339.73	1217.56	874.07	343.49	231.00	.27020
104.304	119	242.28	340.37	1217.75	873.61	344.14	229.30	.27224
105.304	120	244.32	340.99	1217.94	873.15	344.79	227.56	.27421
106.304	121	246.35	341.62	1218.13	872.70	345.43	226.00	.27628
107.304	122	248.39	342.24	1218.32	872.25	346.07	224.40	.27828
108.304	123	250.43	342.85	1218.51	871.80	346.71	222.80	.28027
109.304	124	252.46	343.46	1218.69	871.35	347.34	221.20	.28227
110.304	125	254.50	344.07	1218.88	870.91	347.97	219.50	.28422
111.304	126	256.54	344.68	1219.07	870.47	348.60	218.20	.28625
112.304	127	258.57	345.28	1219.25	870.03	349.22	216.70	.28824
113.304	128	260.61	345.87	1219.43	869.60	349.83	215.20	.29023
114.304	129	262.64	346.46	1219.61	869.16	350.45	213.70	.29222
115.304	130	264.68	347.06	1219.79	868.74	351.06	212.07	.29419
116.304	131	266.72	347.64	1219.97	868.31	351.66	210.90	.29618
117.304	132	268.75	348.23	1220.15	867.88	352.27	209.50	.29816
118.304	133	270.79	348.80	1220.32	867.46	352.86	208.10	.30013
119.304	134	272.82	349.38	1220.50	867.04	353.46	206.70	.30209
120.304	135	274.86	349.95	1220.67	866.62	354.05	205.18	.30406
121.304	136	276.89	350.52	1220.85	866.21	354.64	204.10	.30601
122.304	137	278.93	351.09	1221.02	865.79	355.23	202.80	.30796
123.304	138	280.96	351.75	1221.19	865.38	355.81	201.50	.30990
124.304	139	283.00	352.21	1221.36	864.97	356.39	200.20	.31186
125.304	140	285.04	352.76	1221.53	864.56	356.97	198.78	.31385
126.304	141	287.07	353.32	1221.70	864.16	357.54	197.80	.31586
127.304	142	289.11	353.87	1221.87	863.76	358.11	196.60	.31788
128.304	143	291.15	354.42	1222.04	863.36	358.67	195.40	.31990
129.304	144	293.18	354.96	1222.20	862.96	359.24	194.20	.32190
130.304	145	295.22	355.50	1222.37	862.57	359.80	192.83	.32354
131.304	146	297.25	356.04	1222.53	862.17	360.36	191.90	.32592
132.304	147	299.29	356.57	1222.69	861.78	360.91	190.80	.32794
133.304	148	301.33	357.10	1222.85	861.39	361.46	189.70	.32995
134.304	149	303.36	357.63	1223.02	861.01	362.01	188.60	.33196
135.304	150	305.40	358.16	1223.18	860.62	362.56	187.26	.33315

THE STEAM ENGINE INDICATOR.

The steam engine indicator is now so well known, and so much used, that remarks on its history or construction are unnecessary. From a continuous experience of nearly twelve years with the McNaught, Richards and Thompson indicators, the author feels competent to make a few remarks on the use of this invaluable instrument, and upon the diagram of steam development obtained by it.

The office of the indicator is to furnish a diagram of the action of the steam in the cylinder of an engine during one or more revolutions of the crank ; from which is deduced the following data: Initial pressure in cylinder — piston stroke to cut-off—reduction of pressure from commencement of piston stroke to cut-off—piston stroke to release—terminal pressure—gain in economy due expansion—counter pressure, if engine is worked, non-condensing—vacuum as realized in the cylinder, if engine is worked condensing—piston stroke to exhaust closure, usually reckoned from zero point of stroke, value of cushion—effect of lead, and mean effective pressure on the piston during complete stroke. The indicator diagram, when taken in connection with the mean area, and stroke of piston, and revolution of crank for a given length of time, enables us to ascertain the power developed by engine ; and, when taken in connection with the mean area of piston, piston speed, and ratio of cylinder clearance, enables us to ascertain the steam accounted for by the engine.

The mean power developed by engine compared with the steam delivered by the boilers, furnishes the cost of power in steam ; and when compared with the coal, furnishes the cost of the power in fuel.

The diagram also enables us to determine, with precision, the size of steam and exhaust ports necessary under given conditions—to equalize the valve functions—to measure the loss of pressure between boiler and engine—to measure the loss of vacuum between condenser and cylinder—to determine leaks into, and out of, the cylinder—to determine relative effects of jacketed and unjacketed cylinders—and to determine effects of expansion in one cylinder and in two or more cylinders.

The diagram is frequently used as an exponent of the engine from which it is taken, but it is not always that diagram which, to the observer, looks the most perfect, that represents the best economy.

Experience has shown that other data than the indicator diagrams are necessary to a correct estimate of the economy of performance of an engine.

Although calculated to serve good ends, the steam engine indicator, like the surgeon's knife, should never be applied by unskillful hands.

The cut represents the Thompson indicator, at present the most improved form of the instrument, which, during the past three years, has almost entirely superseded the justly celebrated Richards indicator.

INDICATED H. P. HARRIS-CORLISS ENGINE.

Diam. of cylinder.	Piston speed in feet per min.	*Initial Pressure* 50 *pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE. .10	.15	.20	.25	.30	.35
8	340	4 715	7.878	10 566	12.885	14.905	16.671
10	400	8 680	14.482	19 423	23.686	27.400	30 645
12	450	14.042	23 462	31.466	38.373	44.389	49 646
14	"	19.113	31.934	42 829	52.230	60.418	67.574
15	"	21 939	36.658	49.164	59.955	69 355	77.569
16	500	27 737	46 343	62 154	75 796	87.679	98 066
18	"	35.105	58.654	78.664	95.931	110.970	124.116
20	"	43 340	72.413	97.118	118 432	137.000	153 228
23	"	57 317	95.768	128.439	156 628	181.188	202.647
24	"	62.409	104.275	139 849	170.545	197.283	220.648
26	"	73 243	122 380	164.127	200.152	231.531	258.952
28	"	84.945	141 928	190.348	232.129	268.522	300.324
30	"	97 650	162.929	218.515	266.472	308.250	344 763
32	"	103.248	185.372	248.616	303.184	350.716	382.264
34	"	125 252	209.273	280.671	342.268	395.930	442 829
36	"	140.420	234.616	314.656	383.724	443.880	496.464

Diam. of cylinder.	Piston speed in feet per min.	*Initial Pressure* 60 *pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE. .10	.15	.20	.25	.30	.35
8	340	6.724	10.325	13.426	16.103	18.433	20 470
10	400	12.357	18 981	24.681	29 601	33.885	37.630
12	450	20.020	30 749	39 984	47.954	54.596	60.962
14	"	27 249	41.853	54.422	65.271	74.717	82.975
15	"	31 280	48.044	62.472	74.925	85.769	95.249
16	500	39.544	60 738	78.980	94.723	108.430	120.415
18	"	50 049	76.872	99.960	119.886	137.233	152.403
20	"	61.784	94.904	123.405	148.005	169.425	188.150
23	"	81.716	125.513	163.207	195.740	224.068	248 834
24	"	88 974	136 660	177.705	213.127	243 970	270.934
26	"	104.423	160.387	208.559	250.132	286.332	317.972
28	"	121.094	185 994	241.851	290.064	332.042	368.741
30	"	139.014	213.534	277.661	333.011	381.206	423 340
32	"	158.176	242.952	315 920	378.892	433.720	481.660
34	"	178.555	264.272	356 640	427.734	486 638	543 753
36	"	200.196	317.488	399.840	479.544	548.932	609 612

INDICATED H. P. HARRIS-CORLISS ENGINE.

Diam. of cylinder.	Piston speed in feet per minute.	*Initial Pressure 70 pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE.					
		.10	.15	.20	.25	.30	.35
8	340	8.636	12.772	16.287	19.276	21.962	24.270
10	400	15.875	23.479	29.940	35.515	40.371	44.615
12	450	25.718	38.036	48.504	57.537	65.403	72.278
14	"	35.006	51.783	66.019	78.314	89.018	98.378
15	"	40.183	59.429	75.784	91.992	102.187	112.930
16	500	50.800	75.132	95.807	113.650	129.187	142.771
18	"	64.295	95.091	121.260	143.840	163.504	180.696
20	"	79.377	117.395	149.699	177.578	201.855	223.079
23	"	104.978	155.256	197.979	234.849	266.956	295.025
24	"	114.300	169.047	215.566	255.712	290.671	321.235
26	"	134.146	198.399	252.991	300.109	341.137	377.007
28	"	155.578	230.091	293.413	348.056	395.639	437.240
30	"	178.598	264.139	336.823	399.550	454.174	501.928
32	"	203.200	300.528	383.228	454.600	516.748	571.084
34	"	229.284	339.271	432.630	513.200	583.361	644.698
36	"	257.180	380.364	485.440	575.360	654.016	722.784

Diam of cylinder.	Piston speed in feet per minute.	*Initial Pressure 80 pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE.					
		.10	.15	.20	.25	.30	.35
8	340	10.596	15.219	19.147	22.538	25.490	28.069
10	400	19.478	27.978	35.198	41.430	46.857	51.599
12	450	31.556	45.325	57.021	67.119	75.910	83.593
14	"	42.950	61.692	77.612	91.356	103.321	113.777
15	"	49.303	70.817	89.092	104.869	118.304	130.605
16	500	62.331	89.528	112.631	132.577	149.940	165.116
18	"	78.889	113.310	142.533	167.795	189.771	208.977
20	"	97.390	139.891	175.990	207.152	234.282	257.994
23	"	128.804	185.007	232.750	273.962	309.841	341.200
24	"	140.645	201.438	253.420	298.298	337.365	371.511
26	"	164.596	236.474	297.420	350.090	395.940	436.013
28	"	190.888	274.182	344.937	406.022	459.198	505.661
30	"	219.127	314.755	395.977	466.092	527.134	580.486
32	"	249.324	358.112	450.524	530.308	599.760	660.464
34	"	281.457	404.285	508.611	598.669	677.075	745.602
36	"	315.556	453.240	570.212	671.180	758.084	835.908

INDICATED H. P. HARRIS-CORLISS ENGINE.

Diam. of cylinder.	Piston speed in feet per min.	*Initial Pressure 90 pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE. .10	.15	.20	.25	.30	.35
8	340	12.536	17.666	22.008	25.755	29.018	31.870
10	400	23.045	32.475	40.456	47.345	53.343	58.585
12	450	37.333	52.611	65.542	76.702	86.417	88.576
14	"	50.814	71.609	89.209	104.397	117.622	129.183
15	"	58.331	82.199	102.404	119.839	135.020	148.289
16	500	73.742	103.920	129.461	151.502	170.694	187.473
18	"	93.332	131.525	163.851	191.752	216.040	237.273
20	"	115.223	162.375	202.283	236.728	266.716	292.927
23	"	152.387	214.745	267.522	313.076	352.736	387.400
24	"	165.919	233.820	291.287	340.880	384.061	421.811
26	"	194.728	274.416	341.861	400.073	450.744	495.051
28	"	225.839	318.258	396.478	463.985	522.757	574.142
30	"	259.252	365.344	455.137	532.638	600.111	659.086
32	"	294.968	415.580	517.844	606.008	682.416	749.892
34	"	332.994	469.263	584.597	684.144	770.809	846.559
36	"	373.328	526.100	655.404	767.008	864.160	949.092

Diam. of cylinder.	Piston speed in feet per min.	*Initial Pressure 100 pounds above Atmosphere.* CUT-OFF IN PARTS OF STROKE. .10	.15	.20	.25	.30	.35
8	340	14.517	20.113	24.868	28.973	32.546	35.670
10	400	26.686	36.973	45.714	53.259	59.828	65.572
12	450	43.232	59.899	74.057	86.283	96.925	106.228
14	"	58.843	81.526	100.800	117.441	132.230	144.587
15	"	67.547	93.587	115.709	134.812	151.436	165.974
16	500	85.394	118.315	146.281	170.431	191.452	209.826
18	"	108.081	149.744	185.139	215.704	242.309	265.564
20	"	133.432	184.867	228.570	266.299	299.143	327.853
23	"	176.465	244.489	302.287	352.184	395.621	433.590
24	"	192.136	266.209	329.132	383.470	430.767	472.108
26	"	225.501	312.428	386.278	450.049	505.045	554.076
28	"	261.523	362.343	448.991	521.951	586.327	642.598
30	"	300.222	415.951	514.282	599.173	673.072	737.669
32	"	441.576	473.260	585.124	681.724	765.808	839.304
34	"	385.618	534.275	660.567	796.604	864.523	987.495
36	"	432.324	598.976	740.556	862.816	969.236	1062.256

INDICATED H. P. HARRIS-CORLISS ENGINES.

The development of power for different steam pressures and points of cut-off, is based on the mean effective pressure above the atmosphere. And if it be desired to know the power when engine is worked condensing, under same conditions of initial pressure cut-off, and piston speed; then in every case add to the power in the tables the following values:

8″	cylinder	7.120	H. P.	23″	cylinder	86.558	H. P.
10″	"	13.090	"	24″	"	94.246	"
12″	"	21.206	"	26″	"	110.609	"
14″	"	28.863	"	28″	"	128.280	"
15″	"	33.183	"	30″	"	147.262	"
16″	"	41.887	"	32″	"	167.548	"
18″	"	53.013	"	34″	"	189.150	"
20″	"	65.450	"	36″	"	212.052	"

This table is based upon an assumed vacuum (in the cylinder) of 27 inches corresponding to pres. of 13.25 pounds, to which add .50 pd. counter pressure, which with engine condensing is utilized in mean effective pressure. Suppose a 20″ engine at 500 ft. piston speed, initial pressure 80 pounds and cut-off .20 of piston-stroke, is to be operated condensing: What will be the indicated power? The power above atmosphere by table is 175.990

Add power due vacuum 65.450

241.440 H. P.

HARRIS-CORLISS ENGINES.

Dimensions Cylinder, Piston Speed, and Revolutions.

Cylinder.	Piston Speed.	Revolutions.	Cylinder.	Piston Speed.	Revolutions.
8 × 24	340′	85	20 × 48	500′	62.5
10 × 24	340′	85	20 × 60	500′	50
10 × 30	400′	80	23 × 42	500′	71.43
12 × 30	400′	80	23 × 48	500′	62.5
12 × 36	450′	75	23 × 60	500′	50.
14 × 36	450′	75	24 × 48	500′	62.5
14 × 42	450′	64.3	24 × 60	500′	50
15 × 36	450′	75	26 × 48	500′	62.5
15 × 42	450′	64.3	26 × 60	500′	50
16 × 36	450′	75	28 × 48	500′	62.5
16 × 42	450′	64.3	28 × 60	500′	50
16 × 48	500′	62.5	30 × 60	500′	50
18 × 42	500′	71.43	32 × 60	500′	50
18 × 48	500′	62.5	34 × 60	500′	50
20 × 42	500′	71.43	36 × 60	500′	50

The preceding tables of power are calculated for above piston speeds. The power will be increased or diminished as the piston

speed or revolutions are varied. If the piston of a 20 × 60 engine be increased to 600 ft. or 60 revolutions, then add 20 per cent. to the power as given by the table.

CONDENSATION AND VACUUM.

The absolute pressure of steam is measured from zero, or perfect vacuum, and consists of the pressure indicated by the steam gauge (which is known as pressure above atmosphere), and the pressure of atmosphere as indicated by the barometer. The latter is for all practical purposes a constant quantity for any given locality, and may be roughly taken at 14.5 lbs., corresponding to 29.50 inches of mercury (vacuum gauges are usually graduated to agree with the scale of barometer, and the vacuum is usually stated in inches of mercury). To the steam pressure, as indicated by the gauge, add 14.5 lbs. for total pressure; thus, if the pressure by the gauge is 60 lbs., the total pressure is 74.5 lbs.

By the same token, when the piston moves forward in an engine, the total pressure on steam side at any point in the stroke of piston *is* the pressure above the atmosphere, plus 14.5 lbs., and the total pressure for whole stroke is the mean pressure above the atmosphere, plus 14.5 lbs.; thus, if the mean pressure for whole stroke is 30 lbs., the total mean pressure is 44.5 lbs., and this 44.5 lbs., whether engine is operated condensing or non-condensing, is the variable factor in estimating the load on the engine.

Now, if the engine be operated non-condensing, the 14.5 lbs. (pressure of atmosphere) on steam side of piston is balanced by a like pressure of atmosphere on exhaust side of piston, and its effect is annihilated; but if the engine be operated condensing, a large proportion of the pressure of atmosphere on exhaust side of piston is removed, and an equivalent portion of the pressure of atmosphere on steam side of piston made to do useful work. With well proportioned condensing apparatus, the pressure of atmosphere on exhaust side of piston can be reduced nearly 90 per cent.; in other words, a vacuum in the cylinder (exhaust end) of 13 lbs (26.5 ins.) can be maintained, and this 13 lbs. pressure per square inch of piston is an absolute gain, and should in all cases be utilized.

In a condensing engine, the exhaust is connected with a tight vessel, or chamber termed the condenser (when the condensed steam is to be returned to the boiler as feed water, to the exclusion of the water used in condensing the steam, a surface condenser is used, and when the condensing water is suitable for pumping into boiler, a jet con-

denser is used. Surface condensers are rarely used with land engines, and are not equal in useful effect to jet condensers).

When the exhaust steam enters the condenser, it is intercepted by a spray of cold water, which takes up the sensible and latent heat in the steam and converts it from an elastic vapor to liquid water, and creates a partial vacuum (a perfect vacuum is never formed in steam engine practice, neither is it desirable for the extra economy of the perfect vacuum as compared with the partial vacuum, is neutralized in effect by the extra load on the air pump and diminished temperature of water to the hot well). The vacuum created in the condenser extends to the exhaust end of cylinder, and the moving piston instead of working against an atmospheric resistance of 14.5 lbs. meets a resistance of but 1.5 lbs., the remaining 13 lbs. of atmospheric load having been removed by the vacuum.

The air pump worked by the engine removes the water of condensation, condensing water, air and vapor from the condenser, and delivers into a hot well, from which the water is drawn to feed the boilers. The expense of engine power in working a well proportioned air pump is trifling, and should not be considered in the selection of condensing apparatus. Many cheap condensers have been devised, and some are now in use, the only merit of which (if it be a merit) is that in first cost they are less expensive than the standard condensing apparatus. A favorite form is the siphon condenser, which has been highly successful in injuring many good engines.

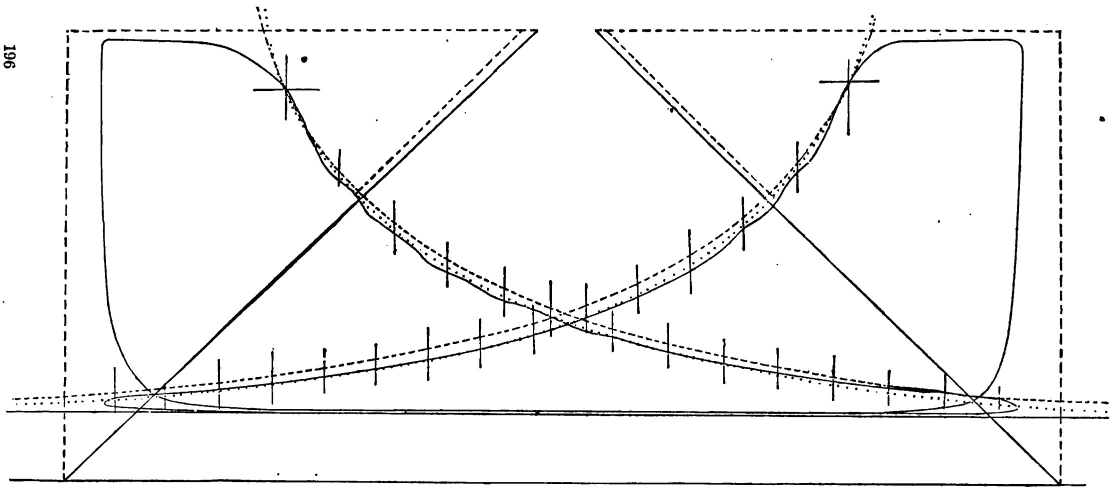
The injector condenser and the ejector condenser have also been tried, with indifferent success, but none of these devices have found favor with steam engineers, from the fact that they can not be depended upon, and are by no means as efficient as the simple condenser and reciprocating air pump.

In adapting engine for maximum economy, care should be had that the terminal pressure, or pressure at release, never falls below atmospheric pressure, otherwise the vacuum will be but partially utilized. In cities where condensing water is obtained from the city mains at a stipulated rate per thousand gallons, careful tests of engine non-condensing should be made before condensing apparatus is added. In nearly every instance it will be found that the cost of condensing water overbalances the gain by the utilization of vacuum; in which case the non-condensing engine will be most economical.

VACUUM IN INCHES OF MERCURY AND POUNDS.*

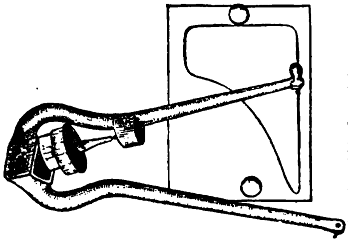
MERCURY.	POUNDS.	MERCURY.	POUNDS.
2 037	1	16 300	8
4.074	2	18.337	9
6.111	3	20.374	10
8 148	4	22.411	11
10 189	5	24.448	12
12 226	6	26.485	13
14 263	7	28 522	14

*Reckoned from atmosphere.



22″ × 36″ HARRIS-CORLISS ENGINE, BETHALTO, ILLS.

THE PLANIMETER.



The planimeter: An instrument for measuring areas of plane surfaces by following the outline of the figure, originated more than fifty years since with M. Oppenkoffer, a Swiss engineer. The ordinary process of measuring plane surfaces having irregular boundaries by dividing the figure into triangles, and computing and aggregating the areas of those triangles, to obtain the areas of the figures is not only tedious, but is liable to serious error where so many independent quantities are to be considered. The planimeter renders the measurement of plane figures of irregular outlines a very easy task, and liable to no appreciable error when worked by one experienced in the use of the instrument.

Notwithstanding the objections to the instrument as invented by Oppenkoffer, nothing better appeared during a period of twenty years' use, when M. Welty, another Swiss engineer, materially improved the planimeter by simplifying its construction; attempting to render it cheaper in cost and more portable.

Five years after the improved planimeter, by Welty, made its appearance, M. Amsler, a professor of mathematics, at Schaffhausen, invented what he termed the polar planimeter, the instrument now largely in use.

The theory of the polar planimeter supposes that every plane surface, without regard to its figure, is composed of an infinite number of small sectors of circles, or of segments* of such sectors, the aggregation of the areas of which is the area of the surface; hence the term "polar planimeter," the pole or center from which the areas of the sectors or the differences of such areas are computed being immovable during the operation of measurement.

The cut represents the Amsler polar planimeter as made by the American Steam Gauge Co., of Boston, one of which the writer has had in daily use for the past six years—many measurements of figures, the areas of which were capable of precise computation by the ordinary methods, having been made by the instrument, to prove its accurracy of performance.

The planimeter furnishes the only exact means of measuring indicator diagrams, omitting Simpson's rule for quadratures: but as the close measurement of a diagram by the latter method requires an amount of time that is simply discouraging to an expert obliged to arrive quickly at results, it is rarely used, except as a beautiful illustration of the mathematical genius of Simpson.

** The use of the term may not be altogether correct, but the cutting line is supposed, in this case, to be an arc of a circle struck from a common polar point.*

ADJUSTMENT OF VALVES.

OF HARRIS-CORLISS ENGINE.

(GEORGE R. BABBITT.)

Radial lines showing the opening or working edges of ports and valves, will be found on the back bonnet side of cylinder, and back end of valves, as follows: For the steam ports, a mark on the cylinder coinciding with that edge of the port towards the end of the cylinder; a mark on the back end of valve coinciding with the edge of valve towards end of cylinder. The lap movement of the steam valve is towards that end of the cylinder in which the valve is located. The exhaust valve covers or works over the opening from the valve chamber into the exhaust chest, and the opening edge is that side of the opening towards the center line of the cylinder, and has its coinciding mark upon the cylinder. The mark on back end of exhaust valve shows its opening edge. The wrist plate is located central between the four ports on the front bonnet side of the cylinder, and has marks on the upper side of its hub showing the extremes of its travel and its center of motion.

To set the valves, place and hold the wrist plate on the center mark, or at the center of vibration, and by the adjusting threads for shortening and lengthening the valve connections, set the exhaust valves at the point of opening, and lap the steam valves from ⅛″ to ⅜″ of an inch, according to size of engine, the less amount for an 8″ cylinder, and the larger amount for a 30″ cylinder, and intermediate sizes in proportion. Now connect the wrist plate and eccentric by the eccentric rod and hook, and, with the eccentric loose upon the shaft, roll it over and note if the wrist plate vibrates to the marks of extreme travel; adjust at the screw and socket in the eccentric rod, to make it vibrate to the marks. Now place the crank upon either dead center, and roll the eccentric enough more than a quarter of a revolution in advance of the crank, observing at this time in which direction it is desired to run the engine shaft), to show an opening of the steam valve nearest the piston of from 1-32 to ⅛ of an inch, according to the speed the engine is to run.

This port opening at the dead center is commonly called lead, and is for the purpose of making an elastic cushion for the piston to rebound from or stop against. High-speed engines require more lead than slow-running engines, other things being equal.

Now tighten securely the set screw in the eccentric, and turn the

engine shaft over in the direction desired to run it, and note if the other steam valve is set relatively the same; if not, adjust by shortening or lengthening its connection.

At a state of rest the weight of the regulator balls rests upon a pin in the side of the regulator column. To adjust the cam rods, have the balls resting upon the stop motion pin; then move and hold the wrist plate to one extreme of its throw, and adjust the cam rod for the steam valve, now wide open, so as to bring the steel cam on the cam collar in contact with the circular limb of the cut-off hook; move the wrist plate to the other extreme of throw, and adjust the other cam rod in the same manner.

To test the correctness of the cut-off, block up the regulator to about its medium height, and with the eccentric connected to wrist plate, roll the engine shaft very slowly in the direction it is to run, and when the cut-off hook is detached by the cam, stop and measure upon the guide the distance traveled by the cross-head; then continue the revolution of the shaft, and note when the other steam valve is tripped. if cut-off is equalized the distance traveled on the guides will be the same; if not, adjust the cut-off rods until the points of cut-off measure alike. The pin in the side of the regulator column upon which the weight of balls rest, is to be removed when the engine is in motion and up to speed, which allows the stop-motion cams to become operative, and stop the engine in case of any breakage of the governor belt, which would allow the engine to run away unless thus guarded against.

AUTOMATIC CUT-OFF AND THROTTLING SLIDE VALVE ENGINE.

Singular as it may seem, there are engine constructors who are yet to learn that the automatic engine is capable of developing a given power at a reduction of 26 to 75 per cent., as compared with the cost of the power by the rank and file of throttling engines.

Under favorable conditions, the loss in economy by the slide valve engine as compared with the cut-off, is nearly 30 per cent.; and a comparison of the performance of slide valve and cut-off engines by test trial, show that 26 per cent. is the minimum saving by automatic cut-off engine.

Comparing the performance of the Harris-Corliss engine at the Cincincinnati Industrial Exposition of 1875 with the performance of several popular slide valve engines, we have as a result the following

relative economy: All the data in the table are from engines operated non-condensing, and (except those designated) at their regular work.

Location.	Date.	Engine.	Class.	Cylinder	Sp'd	St'm p'r hr p'rhp	Relative effi'cy.
Cincinnati,	1875	Harris-Corliss	Auto. c't-off	16″ × 48″	58	23.13	1.0000
"	1877	S. & Co.	Slide-valve	19″ × 54″	68	58.67	0.4943
"	"	J. F. K. & Co.	"	16″ × 30″	60	56.09	0.4124
"	1875	L. & B. Co.	"	9″ × 16″	195	32.34	0.7152*
"	"	B. E. Co.	"	10″ × 14″	210	33.65	0.6814*
Cleveland,	1877	A. & Co.	"	16″ × 29″	70	35.52	0.6512
Dayton,	1874	W. P. C.	"	16″ × 24″	72	66.81	0.3462
Tiffin,	1875	L. & N.	"	16″ × 31″	57	46.35	0.4990
Toledo,	1876	C. & G. C. & Co	"	20″ × 36″	64	51.00	0.4535
Hamilton,	1877	J. H. T. & S.	"	14″ × 20″	104	38.83	0.5957

*Test trials Cincinnati Industrial Exposition, 1875.

DAILY AVERAGE NUMBER OF GALLONS OF WATER PER CAPITA IN THE CITIES NAMED.*

(Dennis Long & Co.)

Washington, D. C	158
New York	100
Brooklyn	50
Philadelphia	55
Baltimore	40
Chicago	75
Boston	60
Albany, N. Y	80
Detroit	83
Jersey City, N. J.	99
Buffalo, N. Y	61
Cleveland	40
Columbus	30
Montreal	55
Toronto	77
London, England	29
Liverpool, "	23
Glasgow, Scotland	50
Edinburg, "	38
Dublin, Ireland	25
Paris, France	28
Tours, "	22
Toulouse, "	26
Lyons, "	20
Leghorn, Italy	30
Berlin, Prussia	20
Hamburg, "	33

* Including water used for manufacturing, fountains, and waste.

SAFETY VALVES.

Let L = length of lever in inches from fulcrum to point of application of weight.
L' = length of lever from fulcrum to center of valve.
L'' = length of lever from fulcrum to its center of gravity.
W = weight of 'P' in pounds.
w = weight of lever in pounds.
w' = weight of valve plug in pounds.
a = area of valve (orifice through seat) in square inches.
p = pressure in pounds per square inch.

$$W = \frac{a\,p - \left(\dfrac{w\,L''}{L'} + w'\right) L'}{L}$$

$$L = \frac{a\,p - \left(\dfrac{w\,L''}{L'} + w'\right) L'}{W}$$

$$p = \frac{\dfrac{W\,L}{L'} + \left(\dfrac{w\,L''}{L'} + w'\right)}{a}$$

Suppose a safety valve in which $a = .442$ sq. inch $L = 18''$ $L' = 2''$ $L'' = 13.875''$ $w = 4$ pounds, and $w' = .25$ pound, what weight of 'P' is required to balance a pressure $p = 1{,}000$ pounds per square inch.

$$W = \frac{.442 \times 1{,}000 - \left(\dfrac{4 \times 13.875}{2} + .25\right) \times 2}{18} = 46 \text{ pounds.}$$

$$L = \frac{.442 \times 1{,}000 - \left(\dfrac{4 \times 13.875}{2} + .25\right) \times 2}{46} = 18 \text{ inches.}$$

$$p = \frac{\dfrac{46 \times 18}{2} + \left(\dfrac{4 \times 13.875}{2} + .25\right)}{.442} = 1{,}000 \text{ pounds per sq. inch.}$$

COMPRESSION.

The following is Mr. Porter's formula for the maximum pressure of compression for steam engines:

Let W = weight of reciprocating parts in pounds.
L = radius of crank in feet.
r = revolutions per second.
n = constant = 1.227.
a = area of piston in square inches.
p = pressure per square inch required.

Then—

$$p = \frac{W\,L\,1.227\,r^2}{a}$$

PILE DRIVING.

Let W = weight of the ram in pounds.
h = fall of the ram in inches.
E = modulus of elasticity of pile.
L = length of pile in inches.
a = sectional area of pile in sq. inches.
s = depth in inches through which pile **was driven by last** blow.
P = maximum load which pile will carry.

Then, according to Rankine—

$$P = \left(\sqrt{\frac{4\,E\,a\,W\,h}{L} + \frac{4\,E^2\,a^2\,s^2}{L^2}}\right) - \frac{2\,E\,a\,s}{L}$$

According to Weisbach, adopting Rankine's form of expression—

$$P = \left(\sqrt{\frac{2\,E\,a\,W\,h}{L} + \frac{E^2\,a^2\,s^2}{L^2}}\right) - \frac{E\,a\,s}{L}$$

And according to Major John Saunders, U. S. A.—

$$P = \frac{W\,h}{3\,s}$$

Data, from Weisbach's illustration. Weight of ram (W), = 2,000 pounds, fall of ram (h), = 72 inches, modulus of elasticity of spruce pile (E), = 1,560,000 pounds, length of pile (L), = 25 × 12 = 300 inches, area of crosssection (a), = 12 × 12 = 144 sq. inches, distance pile was driven by last blow(s), = .2 inch.

And—

$$P = \sqrt{\frac{4 \times 1{,}560{,}000 \times 2{,}000 \times 72}{300} + \frac{4 \times 1{,}560{,}000^2 \times 144^2 \times .2^2}{300^2}}$$

$$- \frac{2 \times 1{,}560{,}000 \times 144 \times .2}{300} = \sqrt{521{,}022{,}390{,}000} - 299{,}522.9 = 422{,}298.9 \text{ pds.}$$

according to Rankine.

$$P = \sqrt{\frac{2 \times 1{,}560{,}000 \times 2{,}000 \times 72}{300} + \frac{1{,}560{,}000^2 \times 144^2 \times .2^2}{300^2}}$$

$$- \frac{1{,}560{,}000 \times 144 \times .2}{300} = \sqrt{238{,}083{,}170{,}000} - 149{,}771.8 = 338{,}161 \text{ pounds, ac-}$$

cording to Weisbach; and—

$$P = \frac{2{,}000 \times 72}{3 \times .2} = \frac{144{,}000}{.6} = 240{,}000 \text{ pounds, according to Major Saun-}$$

ders.

Mr. Trautwine suggests the following for maximum resistance of piles:

$$P = \sqrt[3]{\frac{h}{12}}\,W\,60 = \sqrt[3]{6} \times 2{,}000 \times 60 = 224{,}052 \text{ pounds.}$$

This formula, however, is only applicable when the pile refuses to sink under a given weight and fall of ram.

The author prefers the Weisbach formula, and a factor of safety of 4 to 10, depending upon the value and importance of superstructure.

GENERAL INDEX.

INDEX TO BUSINESS CARDS.

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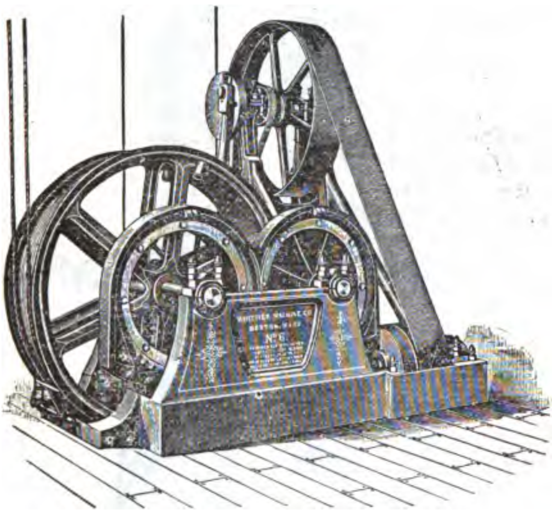

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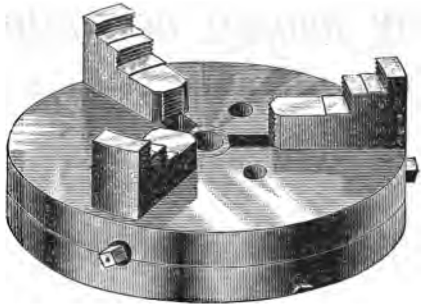

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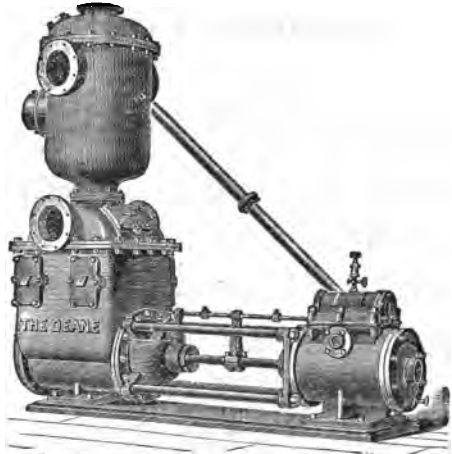
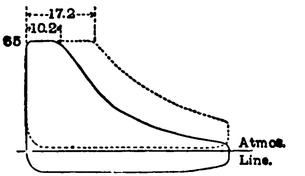
THE DEANE PATENT

INDEPENDENT CONDENSING APPARATUS.

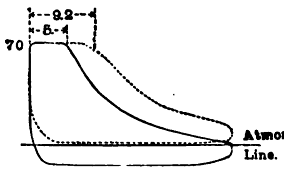


Will save from 20 to 40 per cent of steam used.

The following indicator diagrams illustrate the saving of steam effected by the use of the Deane Condenser.



Above was taken from the cylinder of a "Harris-Corliss" engine, 20 in. by 48 in. The mean effective pressure is 36.7 lbs. per square in. by either diagram. Without the condenser the steam was cut off at 17.2 in., with the condenser at 10.2 in. The steam required to operate the condenser would increase this last to 10.6 in. **The Net Saving** is, therefore, 6.6 in. of steam for every single stroke of the engine, or **38** per cent of that used without condensation.



Above was taken from one cylinder of a pair of Hartford "Buckeye" Engines having 20 in. by 30 in. cylinders. The mean effective pressure is 36.7 lbs. per square in. by either diagram. Without the condenser the steam was cut off at 9.2 in., with the condenser at 5. in. Adding steam required to operate the condenser this last would be increased to 5.5 in. **The Net Saving** is, therefore, 3.7 in. of steam per single stroke of engine, or **40.2** per cent of that used without condensation.

Independent Condensers and Steam Pumps for Every Possible Duty

MANUFACTURED BY

THE DEANE STEAM PUMP CO., Holyoke, Mass.

Boston,	New York,	Chicago.	Philadelphia,
54 Oliver St.	92 and 94 Liberty St.	226 and 228 Lake St.	43 S. 4th St.

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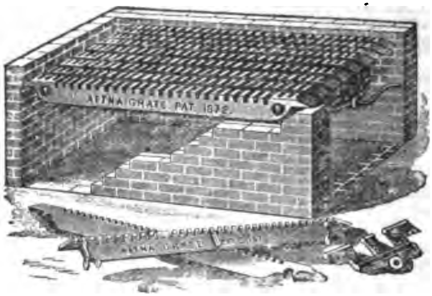
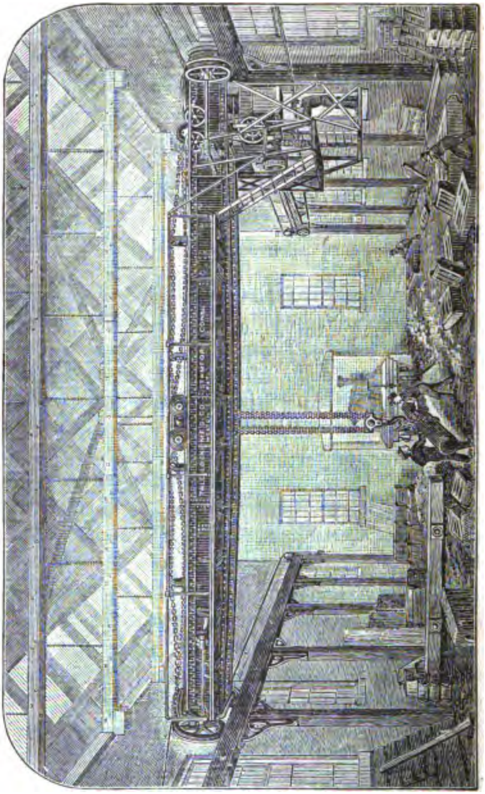

AETNA GRATE PAT 1872

Weston's Patent Cranes,

HAND OR POWER. ANY CAPACITY.

SOLE MAKERS:

Yale Lock Manufacturing Co., Stamford, Conn.



WESTON'S PATENT POWER TRAVELING CRANE.

TRAVELING, JIB, PILLAR, WALKING, AND SPECIAL.

Full specifications and tender submitted on receipt of particulars as to type, capacity, and dimensions of crane required.